1973

Film boiling of benzene, freon 113, freon 11, and normal pentane from horizontal surfaces at atmospheric pressure

Richard Stanley Kistler

Department: Chemical and Biochemical Engineering

FILM BOILING OF BENZENE, FREON 113, FREON 11, AND NORMAL PENTANE FROM HORIZONTAL SURFACES AT ATMOSPHERIC PRESSURE

by

RICHARD STANLEY KISTLER, 1947-

A DISSERTATION

Presented to the Faculty of the Graduate School of the

UNIVERSITY OF MISSOURI-ROLLA

In Partial Fulfillment of the Requirements for the Degree

DOCTOR OF PHILOSOPHY

in

CHEMICAL ENGINEERING

1973

E. Hom Park Jr.
Advisor

Orrin K. Crosser

James W. Johnson

J. L. Gaddy

D J Flanegan

ABSTRACT

A film boiling study was conducted with five different area flat plates at atmospheric pressure for benzene, Freon 113, Freon 11, and n-pentane. The heat transfer surface was horizontal for all tests.

The data were compared to the correlations commonly used for heat transfer. The common correlations were found to be in error for small surface sizes, but adequate for all fluids except benzene for the largest surface, a semi-empirical equation which correlates the available data as a function of surface diameter was derived and discussed.

ACKNOWLEDGEMENTS

The author acknowledges and thanks Dr. E. L. Park, Jr., his advisor, for his help, understanding, and encouragement; Dr. M. R. Strunk, for his support; Mr. Myrlen Troutt for his technical help; his parents, for their support and encouragement; Dr. O. K. Crosser, for his help in writing the thesis; Mr. Gary J. Capone, for his assistance; and The National Science Foundation, The Phillips Petroleum Company, and the Department of Chemical Engineering, University of Missouri-Rolla, for the financial support of this research.

Finally, the author's wife, Bonnie, for too many things to mention here. It is to her this work is dedicated.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

List of Illustrations (continued)

LIST OF TABLES

List of Tables (continued)

List of Tables (continued)

I. INTRODUCTION

This study deals with film boiling from a horizontal surface. The regimes of boiling, including film boiling, were first described in 1934 by Nukiyama (1). The boiling curve, as he described it, was broken into four distinct regions. This curve can best be described by considering a heat transfer surface submerged in a saturated liquid. As the temperature of the surface is raised slightly above the saturation temperature, convective currents circulate the liquid and evaporation takes place at the surface. This region is described as the convective region of heat transfer. As the surface temperature increases further, bubbles form at specific points on the surface. This region is referred to as the nucleate boiling region. As the temperature increases further, more and more points become active until the surface is completely covered with bubbles. This point is called the burn-out point or critical heat flux. This point is of special importance because it is the final point in the nucleate region, where a high heat flux is obtained at a low surface temperature.

As the temperature of the surface increases further, the film formed over the boiling surface collapses and reforms rapidly, causing an increase in the resistance to heat transfer. This region is called the transition region or unstable film boiling region. The efficiency continues to drop until the surface is blanketed by vapor. This point, commonly called the Liedenfrost point, is the minimum point in stable film boiling. As the temperature increases further, the boiling mechanism does not change. This region, called film boiling, is of interest in this investigation.

The purpose of this investigation was to determine the effect of surface diameter on the film boiling curves of benzene, Freon 11, Freon 113, and n-pentane from flat, horizontal copper surfaces at atmospheric pressure.

To the author's knowledge, the effect of surface area on film boiling from a flat, horizontal surface has not been quantitatively determined.

II. LITERATURE REVIEW

Since 1934, when Nukiyama (1) first quantitatively described the four regions of boiling heat transfer as 1) convective heat transfer, 2) nucleate boiling, 3) transition or unstable film boiling, and 4) film boiling, much work has been done in all areas associated with the boiling phenomenon. Because of this fact, only work pertinent to this investigation will be discussed here. For a comprehensive review of boiling heat transfer, the reader is referred to Jakob (2), McAdams (3), Rohsenow (4), and Leppert and Pitts (5). There are also several excellent reviews of film boiling available including Colver (6), Leppert (7), Brentari and Smith (8), and Jordan (9).

The available correlations for predicting the film boiling behavior of fluids from a flat, semi-infinite horizontal surface are based on Taylor-Helmholtz Instability. Taylor (10) first introduced this theory, which is based on a lighter fluid layer lying under a heavier fluid layer in a gravitational field. Assuming that there exists small sinusoidal perturbations at the interface, and that the acceleration is directed from the lighter to the heavier fluid, the disturbance will grow in magnitude until the interface ruptures. When the relative velocities of the fluids are important, this phenomenon is called a Helmholtz Instability (11). This theory postulates that the vapor spacing and frequency of bubble release should be governed solely by the stability of the interface between the liquid and vapor, and the return of liquid to the void created by the departing bubbles is governed by Helmholtz Instability of the two respective innerfaces.

Several authors, including Zuber (12) (13), Lienhard and Wong (14), and Berenson (11) (15) applied Taylor-Helmholtz Instability criterion to determine the minimum heat flux possible in stable film boiling, and the maximum flux possible in nucleate boiling. Since, at the burnout point, in the transition region, and in film boiling, surface effects are assumed negligible (that is, there is no solid-liquid contact), these phenomenons should be governed solely by the hydrodynamic considerations of Taylor-Helmholtz Instability.

Berenson (11) (15) first extended Taylor-Helmholtz Instability criterion to the prediction of film boiling from a flat surface. The resulting equation is:

$$h = 0.425 \left\{ \frac{K^3 \Delta h \rho g (\rho_e - \rho)}{\mu \Delta T \sqrt{\frac{g_c \sigma}{g(\rho_e - \rho)}}} \right\}_f^{\frac{1}{4}} \qquad .....1$$

This equation is identical to Bromley's equation (16) for cylinders except that the characteristic tube diameter is replaced with

$$\sqrt{\frac{g_c \sigma}{g(\rho_e - \rho)}} \quad ,$$

commonly called Laplace's reference length. Berenson also showed, through the application of Taylor-Helmholtz Instability, that the "most dangerous wavelength" (when the vapor bubble departs from the interface) is given by:

$$\lambda_d = 2\pi \sqrt{\frac{3g_c\sigma}{g(\rho_e-\rho)}} \qquad .....2$$

All values for this equation are evaluated at the mean temperature between the surface and liquid temperatures.

Several authors have proposed modifications to Berenson's equation. Ruckenstein (17) multiplied Berenson's equation by a complex constant factor to take into account the radius of a cavity as well as the radius of prominence. Ragsdale (18) (19) suggested changing the constant slightly. Hamil and Baumeister (20), in treating film boiling from a horizontal surface as a optimal boundary value process, suggested modifying the latent heat to account for sensible heat effects. Kermode and Zemaitis (21) modified Berenson's equation by changing the constants slightly and multiplying by the ratio of the liquid viscosity to the vapor viscosity and the ratio of the liquid density to the vapor density each raised to a different power.

Chang (21) applied wave theory to film boiling from a flat, horizontal plate. His final expression is:

$$h = 0.234 \left\{\frac{K^2 g\rho(\rho_e-\rho)\lambda}{\mu_v \Delta T}\right\}_f^{1/3} \qquad .....3$$

Frederking (23) correlated film boiling for nitrogen and helium from a small (0.187 by 0.250 inch) copper surface viewing a saturated liquid through a short passage. His final expression was:

$$h = 0.2 \left\{\frac{K^2 g\rho(\rho_e-\rho)\lambda'}{\mu_v \Delta T}\right\}_f^{1/3} \qquad .....4$$

This equation also fits several organic fluids well. The latent heat of vaporization is corrected for sensible heat effects, and vapor properties are evaluated at an average temperature.

In addition to these theoretical studies, several investigators have studied the behavior of organic fluids in film boiling at several different pressures. Kermode and Zemaitis (21) studied the behavior of benzene, n-hexane, and methanol at atmospheric pressure. They used a five by eight inch copper block which formed the base of the boiling vessel. The block was heated with cartridge heaters, and transite was used to minimize heat losses.

Heath and Costello (24) studied ethanol, n-pentane, and Freon 113 at high surface temperatures at atmospheric pressure. Two platinum elements, back to back, one and one-half inches wide by six inches long and 0.005 inches thick, were mounted in a centrifuge and boiling took place at different accelerations.

Hosler and Westwater (25) studied Freon 11 and water at atmospheric pressure from a eight by eight inch aluminum plate three inches thick. Heat was supplied by the combustion of natural gas, and a fluid depth of one-half inch was used after the investigation of heights of up to 2.12 inches showed no influence of head over the heater. Kautzky and Westwater (26), using the same apparatus, studied the film boiling of carbon tetrachloride and Freon 113 and their mixtures at atmospheric pressure.

Kesselring, Rosche and Bankoff (27) studied Freon 113 at atmospheric pressure using flattened stainless steel tubes covered with a one-eight inch thick epoxy coat as insulation on the tube bottom. All tubes were 4.75 inches long, and surface widths of one-quarter, one-half, and one inch were used. Heat was supplied by condensing steam.

Ragsdale and Sauer (18) (19) studied Freon 11 at atmospheric pressure. The heating elements were four inch long strips of Kanthal A1 and Inconel 600, one-half, one and two inches wide. Different surface treatments were studied including sandblasting and chrome plating. Power was supplied by a welder, and the boiling chamber was open to the atmosphere.

III. EXPERIMENTAL EQUIPMENT

The equipment used in this investigation can be classified into five systems: A) flat plate heating device, B) boiling and condensing system, C) pressure control system, D) electrical power system, and E) temperature measurement system.

A. Flat Plate Heating Device

The heater core, Figure 1, was constructed from copper rod four inches in diameter and five inches long. Six 0.625 inch diameter holes, four and one-half inches deep were drilled for the heating elements. The cartridge heating elements, General Electric catalog number 7C934A101, rated 550 watts at 120 volts, were 0.625 inches in diameter and four inches long.

The heater core was bolted inside a five inch long section of schedule forty steel pipe, four inch nominal size. A four inch floor flange on the base of the pipe, turned down to an outer diameter of 5.60 inches, was bolted to the main support flange by three stove bolts. These bolts were insulated from the heater core by one inch thick Teflon spacers.

The main support flange was formed from a six inch cast blind pipe flange, turned down to 0.750 inches thick, with a four and one-half inch diameter hole in the center. This allowed the surface element to pass through the flange into the boiling chamber. The flange was supported by three eleven inch sections of three-quarters inch pipe.

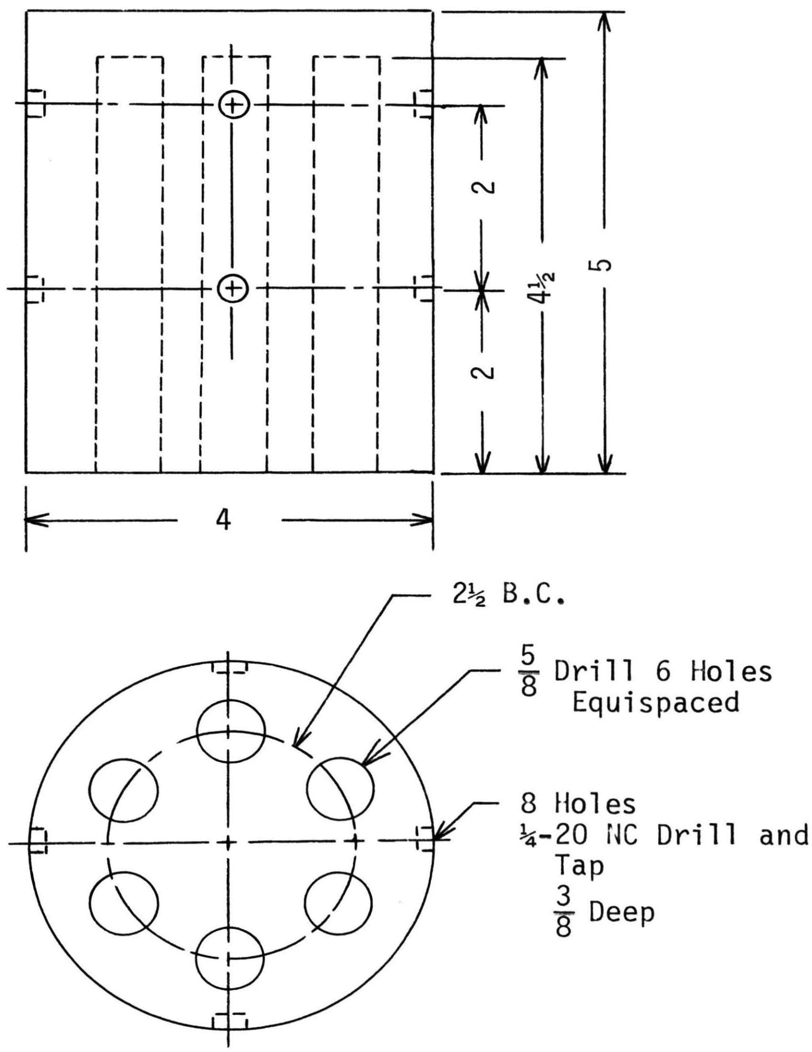
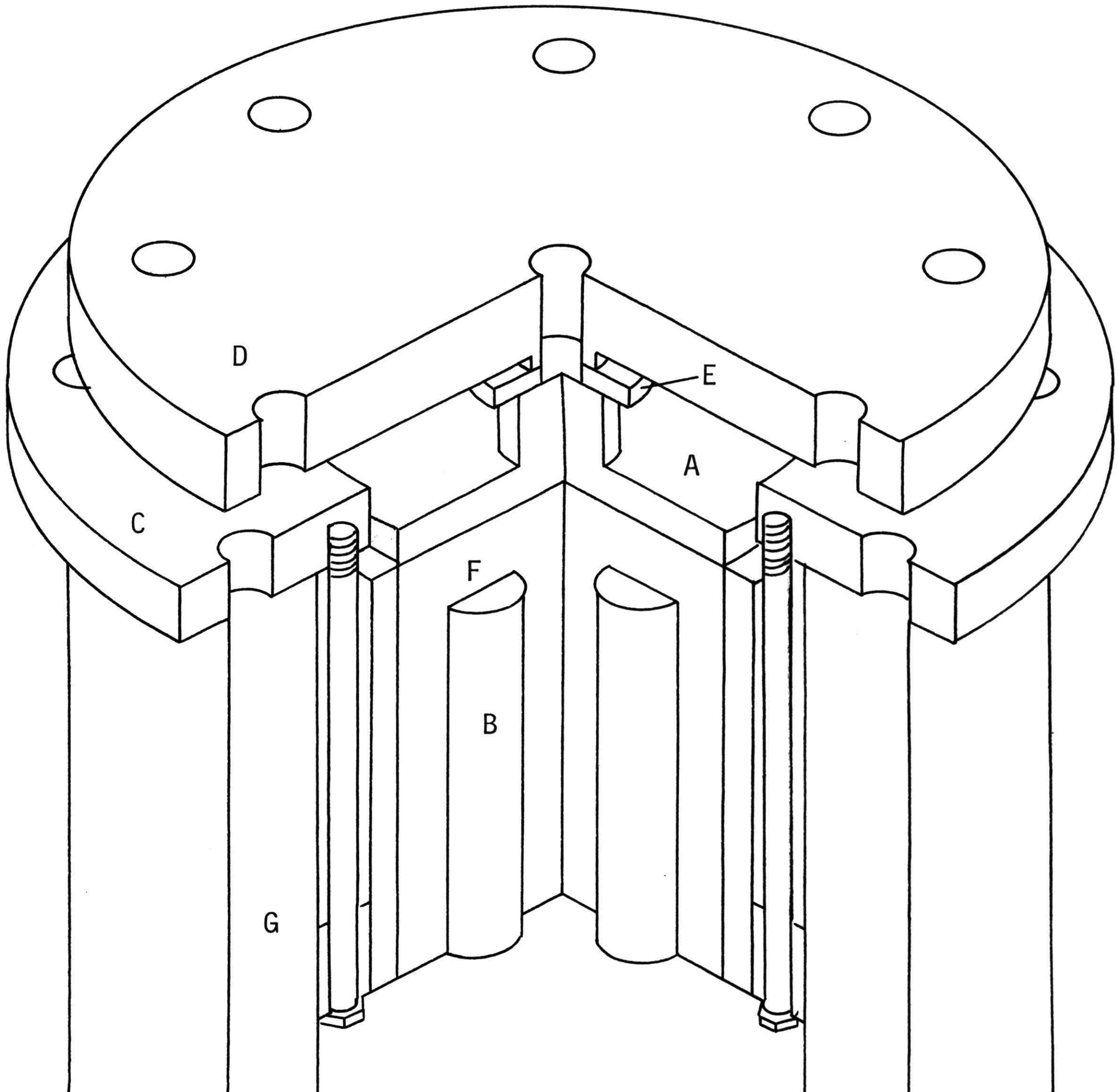


Figure 1: Heater Core



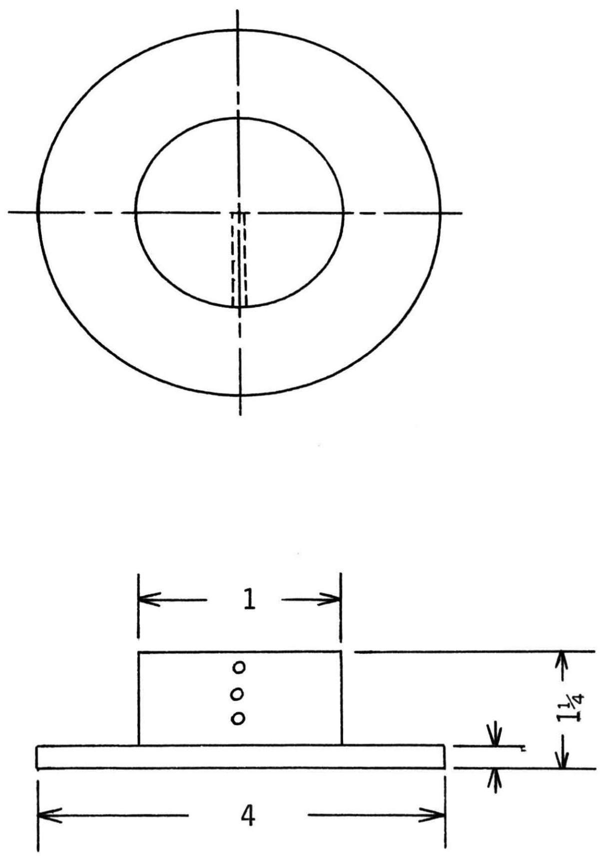
A - Surface Element
B - Cartridge Heating Elements
C - Support Flange
D - Boiling Flange
E - Surface Gasket
F - Heater Core
G - Insulation

Figure 2: Flat Plate Heating Device

Heat losses from the core were minimized through the use of insulation. An eleven inch long section of Carrytemp 1500 five inch pipe insulation one and one-half inches thick was used for this purpose. The air space below the heating element and inside the insulation was filled with rock wool insulation held in place by a Teflon cover. The insulation was held to the main support flange by asbestos covered Chromel A wire. The heating apparatus is shown in Figure 2.

Three different surface elements, Figure 3, with diameters of 1.000, 3.250, and 3.850 inches were used. The 1.000 inch surface was used for test elements of less than one inch in diameter. The 3.850 inch diameter surface was used for the largest test element, and the 3.250 inch surface was used for all other test elements. The 1.000 and 3.250 inch diameter surface elements were 1.250 inches tall, while the 3.850 inch diameter surface element was 0.875 inches tall. There were three thermocouple wells in the 1.000 and 3.850 inch diameter surface elements, while the 3.250 inch diameter surface element had four. The location of the flux thermocouple wells is given in Table I. After the thermocouples were silver soldered they were soldered in the wells. Each thermocouple well was filled with Euctectic number 155 solder. The flux thermocouples were 30 AWG chromel-alumel, Omega Engineering type AHG-K-30.

The floor of the boiling chamber was formed by a four inch pipe forged steel blind flange. A hole of the proper diameter through the center of the flange allowed the surface element to view the boiling chamber. The surface diameters used for this investigation



All Thermocouple Holes = $\frac{1}{16}$ Drill

Figure 3: Surface Elements

TABLE I

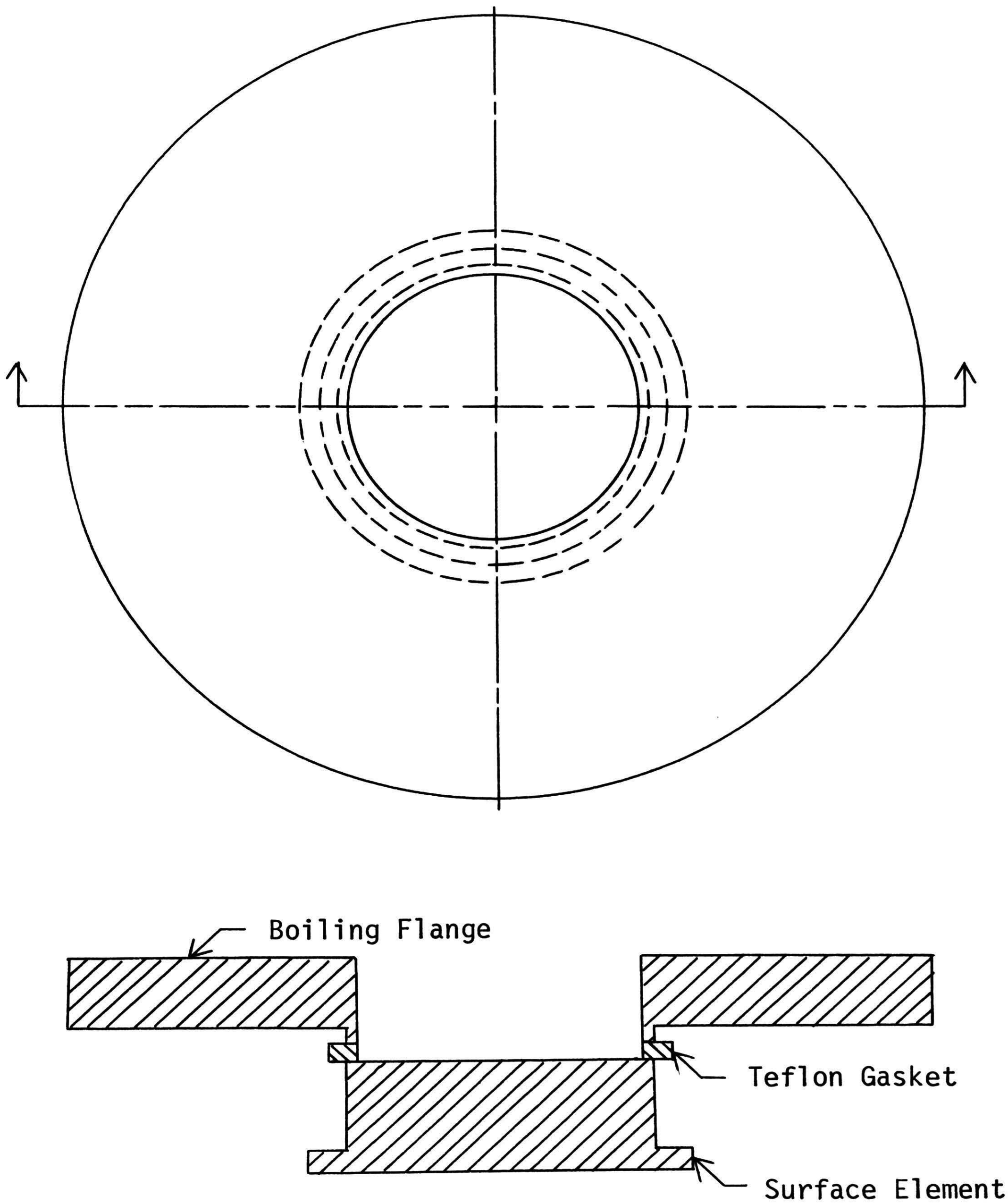
Flux Thermocouple Location

Surface Diameter (inches)	Thermocouple Location (inches from surface) 1	2	3	4
1.000	.125	.374	.884	--
1.250	.132	.373	.623	.871
3.850	.212	.355	.483	--

were 0.4375, 1.016, 2.045, 3.025, and 3.630 inches. The boiling flange was insulated from the surface element by a 0.1875 inch thick Teflon gasket. The flange was machined so that the total path length from the surface to the boiling chamber was one inch. Proper sealing of the surface element against fluid loss was insured by three bolts passing through the main support flange. These three bolts were insulated from the main support flange by one-half inch thick Teflon spacers. These bolts allowed a compression seal on the surface element. Detail of the surface sealing system is shown in Figure 4.

B. Boiling and Condensing System

The boiling chamber, Figure 5, was formed by a six inch diameter section of Pyrex Double Tough glass pipe one foot long. The top of the boiling chamber was formed by a one-half inch thick aluminum plate. The chamber was sealed at the top by a Teflon over Neoprene rubber gasket and by a Teflon over quilted asbestos gasket at the bottom. A General Electric cartridge heating element, catalog



Scale 1"=2"

Figure 4: Detail of Surface Sealing System

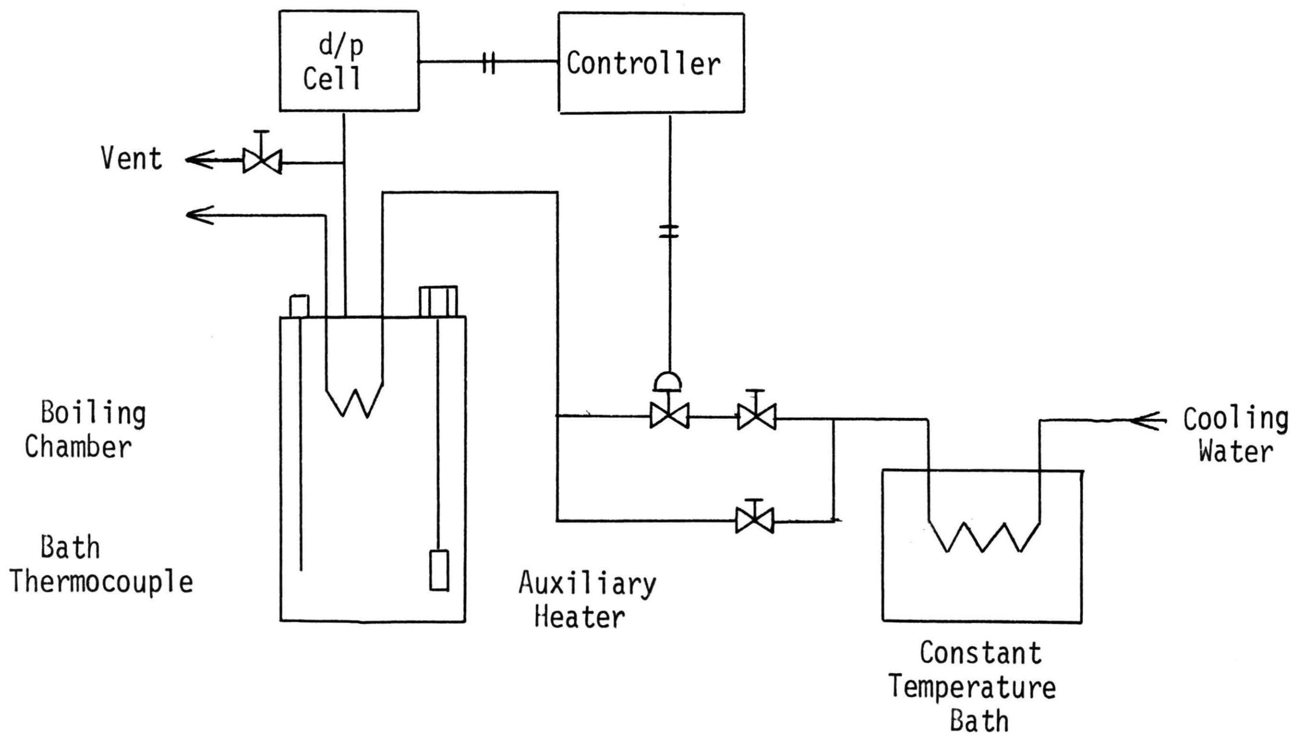


Figure 5: Boiling Chamber and Pressure Control System

number 7C912A101, rated 165 watts at 120 volts, was used as an auxiliary bath heater. The leads for the auxiliary heater passed into the boiling chamber through a Conax gland, type PL-12-B-2. The condensing coil was formed from two and one-half feet of one-quarter inch outer diameter copper tubing wound on a two inch form. The coil passed into the boiling chamber through drilled out Swagelok fittings.

System pressure was monitored by an Aschroft model 1850 pressure gauge with a range of eighty inches of water full scale. One additional tap on the top of the chamber was used when filling and emptying the boiling chamber.

After the boiling chamber was wrapped with a three inch thick bale of fiberglass matt, a twenty-eight inch long section of Carrytemp 1500 ten inch pipe insulation was placed around the heating apparatus. The heating apparatus was then placed in a wooden box, twenty-six by thirty-two by thirty-six inches deep, and the remainder of the box was filled with rock wool. The top of the boiling apparatus was then covered with approximately one foot of rock wool.

C. Pressure Control System

During each of the boiling tests, the system pressure was connected to the high pressure side of a Foxboro model 12A d/p cell. The low pressure side of the cell was vented to the atmosphere. The pneumatic signal transmitted by the d/p cell was sent to a Honeywell pneumatic controller, model 702-G-N-92-III-65. The controller-d/p cell system was calibrated so that a one pound per square inch pressure on the high pressure side of the d/p cell produced full deflection on the controller scale.

The pneumatic output of the controller operated an air-to-open control valve on the water line to the condensing coil. The control valve used was a Research Controls, Inc., type 7566SD-ST-RF equipped with trim N. To minimize pressure cycling, two valves, Hoke type C-32-B, were used in the water line. One valve was placed on the up-stream side of the control valve, while the other was used as a bypass valve around the control valve.

Since the saturation temperatures of the test fluids varied greatly, a constant temperature bath was employed to adjust the inlet water temperature. A CRC model 576010 constant temperature control was used to control the temperature of the bath, and the inlet water was first passed through fifty feet of copper tubing in the bath before being led to the condensing coil. The system pressure was thus controlled by the condenser water flow rate.

D. Electrical Power System

Two Sorenson DC60-40A direct current power supplies, wired in series, were used to supply the power to the heating apparatus. Total output power, as wired, was 4800 watts.

Output current was monitored by a Weston ammeter, calibrated to ± 0.5 percent full scale. The meter was used on the ten amps full scale range. Output voltage was monitored by a Digatec model 201 direct current voltmeter with an accuracy of ± 0.05 percent full scale. The voltmeter was used on the 100 volts full scale range.

E. Temperature Measurement System

In addition to the surface thermocouples and the bath thermocouple, other thermocouples were placed as follows: inlet and outlet cooling water stream, outside the heater base insulation, outside and inside the outer insulation top and bottom, on the edge of the main support flange, and on the bottom of the flange forming the bottom of the cooling chamber. Thermocouple location and number are given in Figure 6. For the three intermediate surface sizes, thermocouple number four is the fourth thermal flux thermocouple, while for the smallest and largest surface it is the bottom temperature of the boiling flange. All thermocouples except numbers nineteen and twenty were 30 AWG chromel-alumel, Omega Engineering type AHG-K-30. The other thermocouples were 24 AWG chromel-alumel, Omega Engineering type GG-K-24. All thermocouple junctions were formed by silver soldering the two twisted wires together.

Thermocouples one through eight were led to the thermocouple switch, a West Instruments 23 position model, through an Amphenol connector, model 26-4100-16P. All other thermocouples were wired directly to the thermocouple switch. Thermocouple compensation for room temperature was accomplished by a West Instruments model AC-II reference junction.

Thermocouple output was monitored on a Digatec model 268 digital voltmeter on the twenty millivolt full scale range. Accuracy was ± 0.05 percent full scale. The voltmeter was connected to a Digatec model 691 digital printer which was triggered automatically by a mechanical drive at three minute intervals.

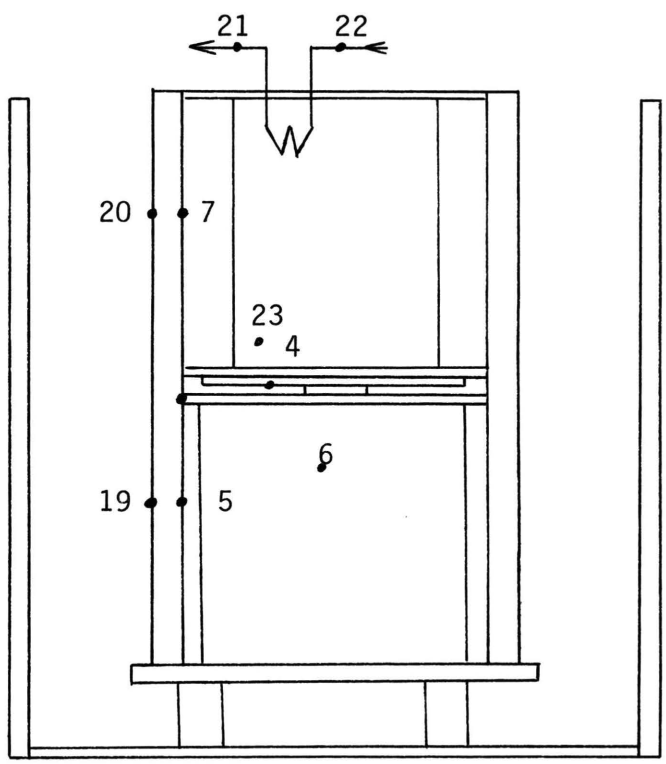


Figure 6: Heat Loss Thermocouple Location

IV. EXPERIMENTAL PROCEDURE

The experimental procedure followed in this investigation is specified in the following sections: A) Startup, B) Initial Film Boiling Tests, C) Final Film Boiling Tests, and D) Heat Loss Tests.

A. Startup

Before the boiling apparatus was assembled, the surface was first sanded with four hundred and six hundred grit emery paper, then fine crocus cloth. The surface was then cleaned with a soft cotton cloth and acetone. The boiling apparatus was then placed in the box and covered with insulation. There was six to twelve inches of insulation covering the boiling apparatus at all points.

After about 20 minutes when all instruments were warmed up, approximately one-half gallon of the test fluid, at ambient temperature, was introduced into the boiling chamber. Introduction of one-half gallon of test fluid gave a fluid height of five inches above the boiling surface. Several investigators (25) (26) (28) have reported that varying the liquid height between one-half and six inches has no effect on the boiling characteristics.

Maximum power was then applied to the heating element. The combination of a large step power input and sub-cooled liquid allowed the surface to go into film boiling very quickly, generally in less than two minutes.

After the surface was in film boiling, the constant temperature bath was set at approximately twenty degrees Fahrenheit below the saturation temperature, and the system was vented several times to purge non-condensable gasses. An ice bath was used in place of the constant temperature bath when Freon 11 was being tested. The auxiliary heater was used only to bring the pool to saturation temperature when benzene was being tested. After saturation pressure and temperature were reached, the auxiliary power was shut off.

After the atmospheric pressure was determined to 0.1 millimeters of mercury, the pressure controller-recorder was set to the proper level to assure a pressure of one atmosphere in the boiling chamber, and the bypass and throttling valves on the water line were adjusted to reduce pressure cycling below ± 0.01 psia.

The power input level was then set so that the temperature of the surface, which was defined as the temperature of the thermocouple closest to the surface, would stabilize in the film boiling region at a low flux level.

The benzene used was a Fisher Certified reagent, with purity exceeding 99 percent. The Freon 11 used was commercial refrigeration grade, manufactured by DuPont. Analysis with a Varian 1520 gas chromatograph indicated that the purity was greater than 99 percent. The Freon 113 used was 99 percent pure, manufactured by Matheson Gas Products, while the n-pentane, manufactured by MCB Industries, Inc., was 98 percent pure. Analysis of these two fluids with a Varian 1520 gas chromatograph showed purity levels greater than 99 percent for both fluids.

B. Initial Film Boiling Tests

All film boiling tests were conducted in a systematic manner. Initially, all data was taken by increasing the power level after the surface temperature had stabilized at the lowest flux level, and allowing the surface temperature to approach steady state. The throttling and bypass valves on the cooling water line were reset to reduce the pressure cycling. Steady state was defined as no change larger than .05 degrees Fahrenheit in surface temperature after ten minutes elapsed time. This procedure generally required three to four hours. The surface temperature was increased fifty to one hundred degrees Fahrenheit for each change in power setting.

After a surface temperature of five hundred to six hundred degrees Fahrenheit was obtained, the above procedures were reversed and several points were taken going back down the curve. These points usually took five to seven hours to reach steady state because of the time required to cool the insulation surrounding the boiling apparatus. This procedure showed no hysteresis effects in initial runs, therefore the descending points were omitted in following tests.

Figure 7 shows the four preliminary runs taken with n-pentane from a 3.025 inch diameter surface. This data is typical of all data taken during this investigation. For these four curves, the data lies within a ten percent region of the mean average represented on the graph. Also, the points taken while descending the film boiling curve are well within the ten percent limitations, showing minimal hysteresis effects.

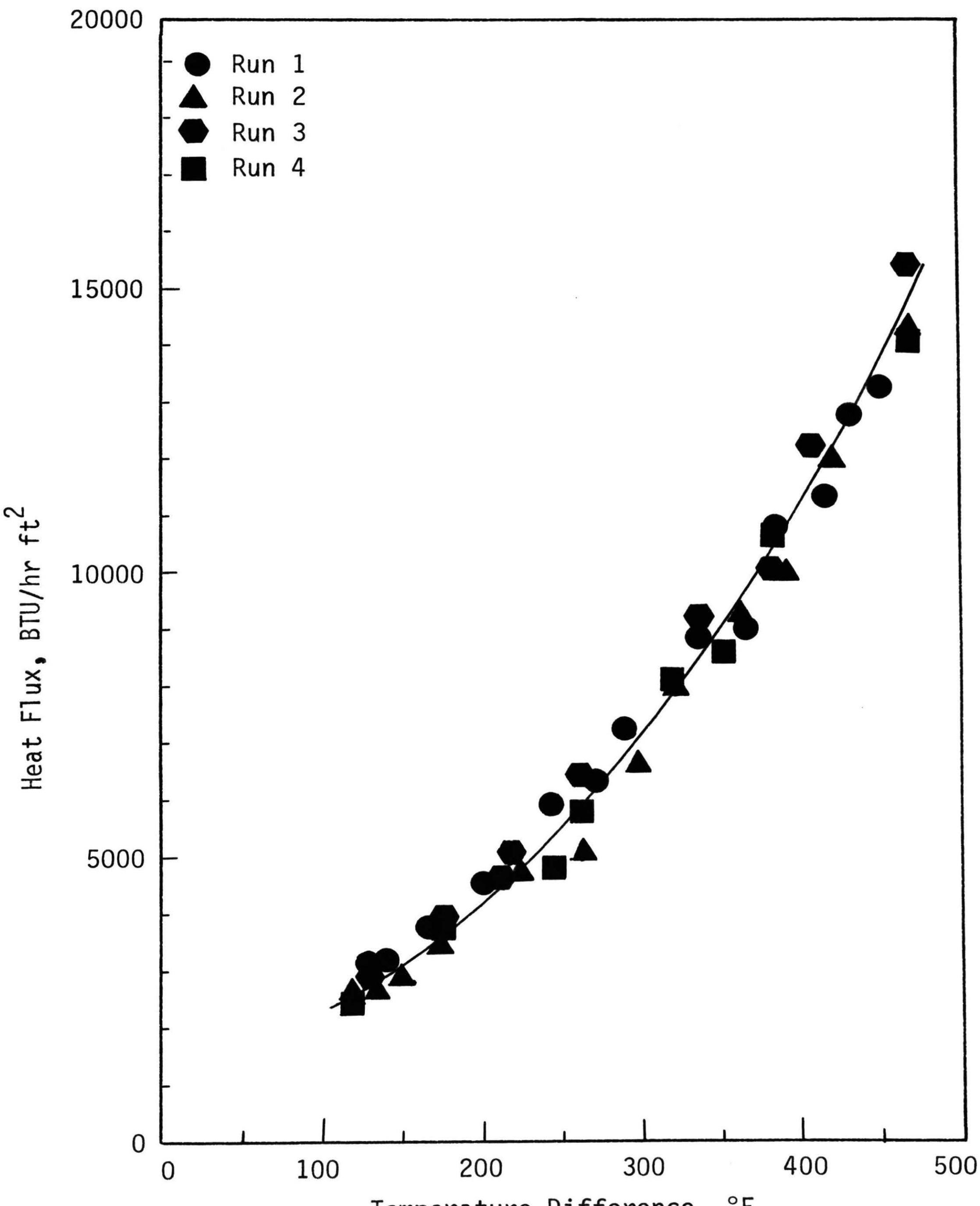


Figure 7: Film Boiling of n-Pentane from a 3.025 Inch Diameter Surface

Visual observation of the preliminary tests verified that the surface was covered with a vapor blanket, indicating that film boiling was taking place. The bubbles departed at specific points from the interface, in regular patterns. However, the points of departure did vary slightly and were not fixed at specific points over a period of time. This pattern is in complete agreement with the work of Holser and Westwater (24).

C. Final Film Boiling Tests

After the preliminary tests were concluded, additional loss thermocouples, the constant temperature bath and cooling water thermocouples were added. The testing procedure was also altered. After the surface temperature had stabilized at the initial point, the power level was increased to approximately five hundred watts. The surface temperature was monitored until an increase of seventy to one hundred degrees Fahrenheit was obtained, then the power was lowered to approximately the proper level. The surface temperature was then monitored for approximately fifteen minutes, until a trend was established. The power level was then increased or decreased, as required, at the rate of one tenth of a volt every three minutes until the surface temperature had not varied more than 0.001 millivolts (approximately 0.05 degrees Fahrenheit) over a nine minute period. Using this method, the average time per data point was one and one-half hours.

When the power level was changed, the water temperature in the constant temperature bath was lowered slightly, and the valves on the cooling water line were adjusted to minimize pressure cycling. Pressure cycling was typically ± 0.01 psia, with a maximum of ± 0.04 psia.

D. Heat Loss Test

Due to the materials of construction of the heating apparatus, fairly high heat losses were encountered during all runs. Several attempts were made to close the heat balance based on the heat loss through the insulation and the heat removed by the condenser water. Also, superheating of one to ten degrees Fahrenheit was observed on the bottom of the boiling flange for each run.

Calculated heat losses through the insulation were not accurate because gradients did exist around the insulation, as it was warmer near the seams where the insulation was assembled than at the center where the temperatures were taken. Based on the above data, the heat balance closed to within 28.7 percent average, with a maximum error of 60.6 percent. Due to the inability to close the heat balance and because of the superheat on the bottom of the boiling flange, additional loss runs were taken.

During these runs, the surface was covered with a 0.1875 inches thick Teflon gasket to prevent any heat loss through the surface. Since all other variables remained constant, the energy required to hold the heater core at a specific level was equal to the heat lost at that particular temperature. At lower core temperatures, it was necessary to use the auxiliary heater to maintain saturation temperature. At higher temperatures, the auxiliary heater was not used.

One run was taken with each fluid using the 0.4375 inch diameter surface, and runs were taken for two fluids, Freon 113 and n-pentane, with the largest surface, 3.630 inches in diameter. The results are shown in Figure 8. Since the values obtained for Freon 113 and n-pentane from the larger surface agreed with the data taken from the smaller surface, no additional loss runs were taken. Each set of loss data was fitted with a least squares line for use in computing the losses at different power levels. These constants as well as the mean percent error for each fit are given in Table II.

TABLE II

Correlation of Heat Loss Data

$$Q_{LOST} = A_0 + A_1 (T_{CORE})$$

Fluid	A_0	A_1	Average Deviation Percent	Maximum Deviation Percent
Benzene	-141.90	1.128	0.87	1.93
n-Pentane	-177.92	1.317	0.66	1.92
Freon 113	- 95.61	1.095	0.65	1.63
Freon 11	-164.94	1.363	0.99	2.76

The description of the experimental apparatus shows that the precision in the measurements of the power input, for 3 inch diameter heating area allow Q/A to be measured to within about ± 0.1 percent at a heat flux level of seven thousand BTU/hour square feet,

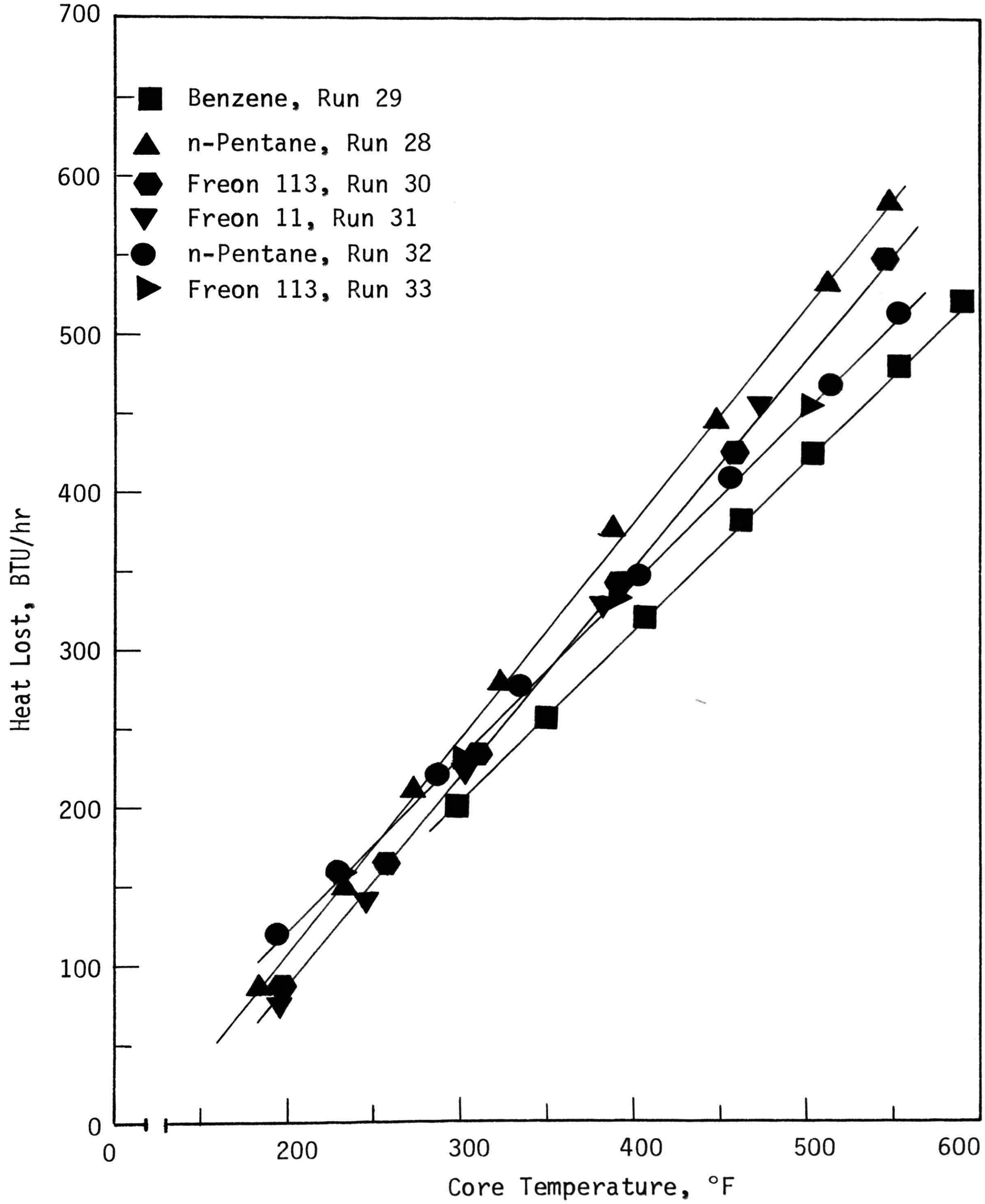


Figure 8: Heat Losses

and a temperature difference of two hundred degrees Fahrenheit to within 0.05 degrees Fahrenheit. This means that the maximum possible scatter in the experimental data should be 0.1 percent in Q/A for this area. However, the uncertainty introduced by heat losses reduces the precision to the order of fifteen percent for this case.

V. RESULTS

The data of this investigation, in the sequence taken, are presented in Appendix A. For this study, the system was operated at a single pressure, 14.7 psia, and only saturated pool boiling was studied.

The results of this investigation are presented in Figures 9 through 13. In these figures, the heat flux is plotted as a function of the difference between the surface temperature and saturation temperature of the boiling liquid at a constant diameter.

In general, benzene proved to be the most difficult fluid to boil, demanding a higher flux level to maintain a given temperature difference. The remaining fluids, n-pentane, Freon 113, and Freon 11, required a progressively lower flux to maintain a given temperature difference.

Two trends are evident from this data. First, as the temperature difference increased, the heat flux also increases. The shape of these curves, at a constant diameter, is approximately the same for each of the four test fluids at temperature differences greater than two hundred degrees Fahrenheit. Second, as the diameter decreases, for surfaces smaller than 3.025 inches, the heat flux increases. Both trends agree with the data presented by Ragsdale and Sauer (18) (19) as well as that of Kesselring, et al., (27) and Kistler, et al., (29).

Maximum scatter observed in the data, as indicated by the data for n-pentane in Figures 10 and 11, is approximately ten percent.

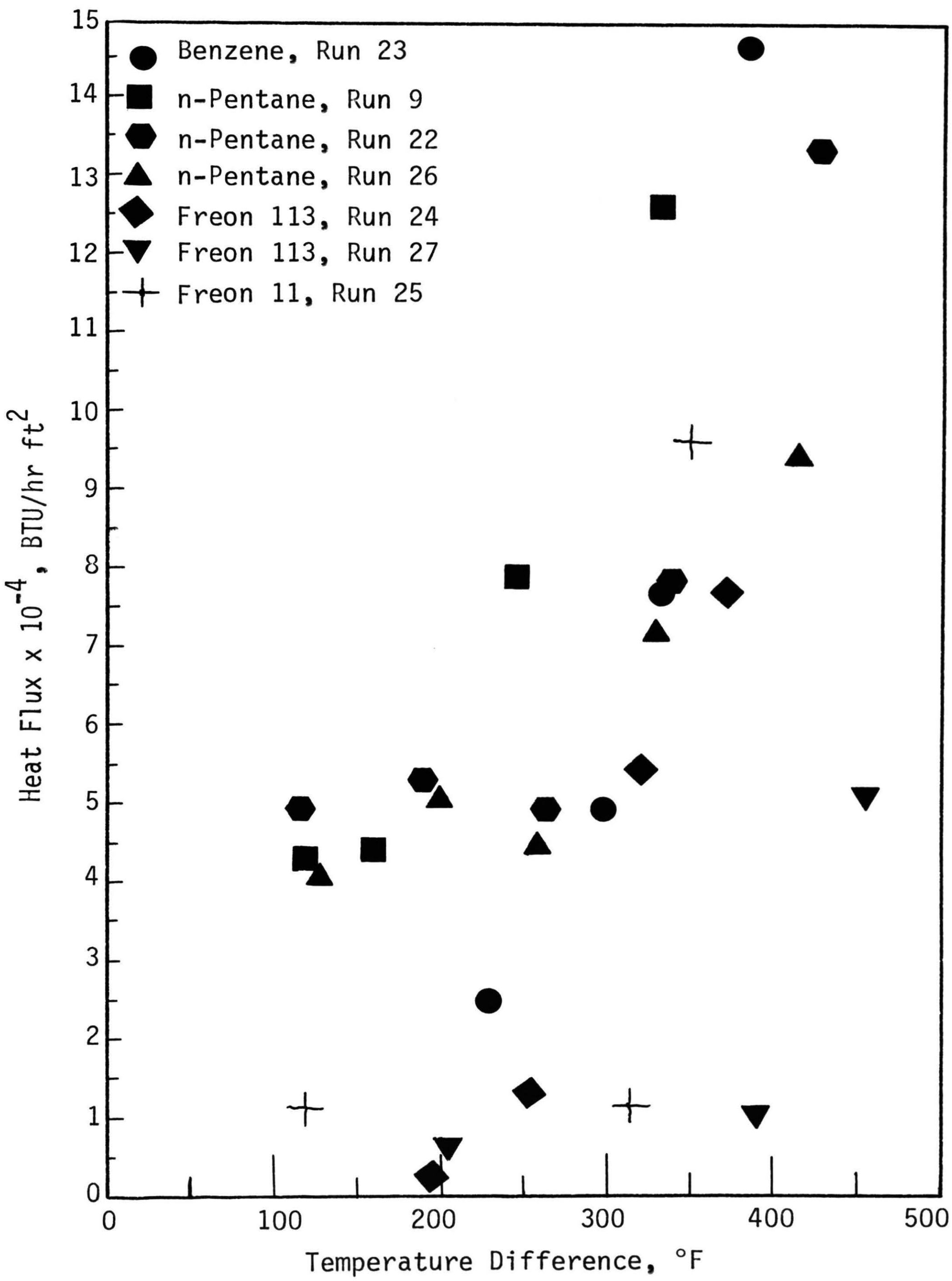


Figure 9: Film Boiling from a 0.4375 Inch Diameter Surface

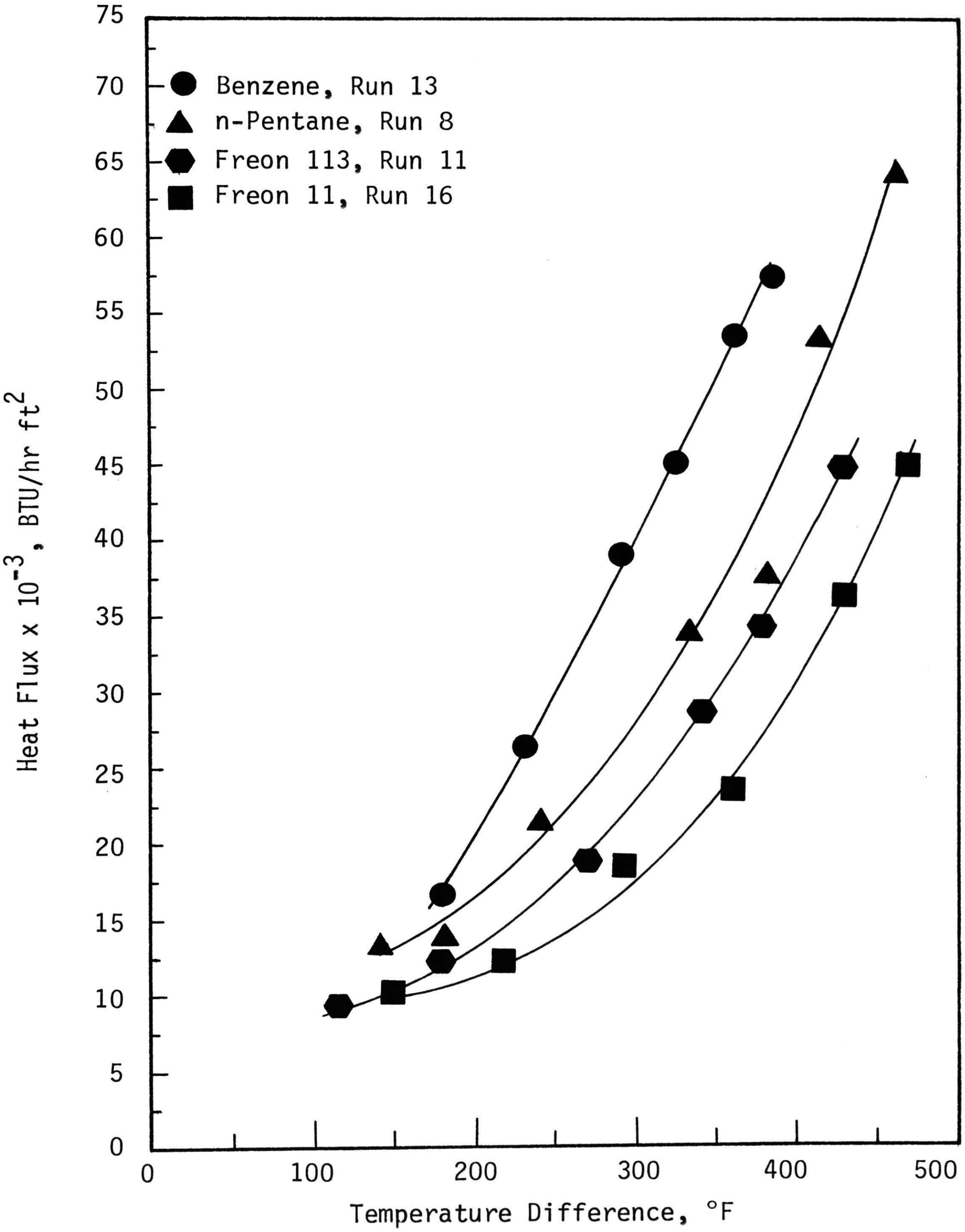


Figure 10: Film Boiling From a 1.016 Inch Diameter Surface

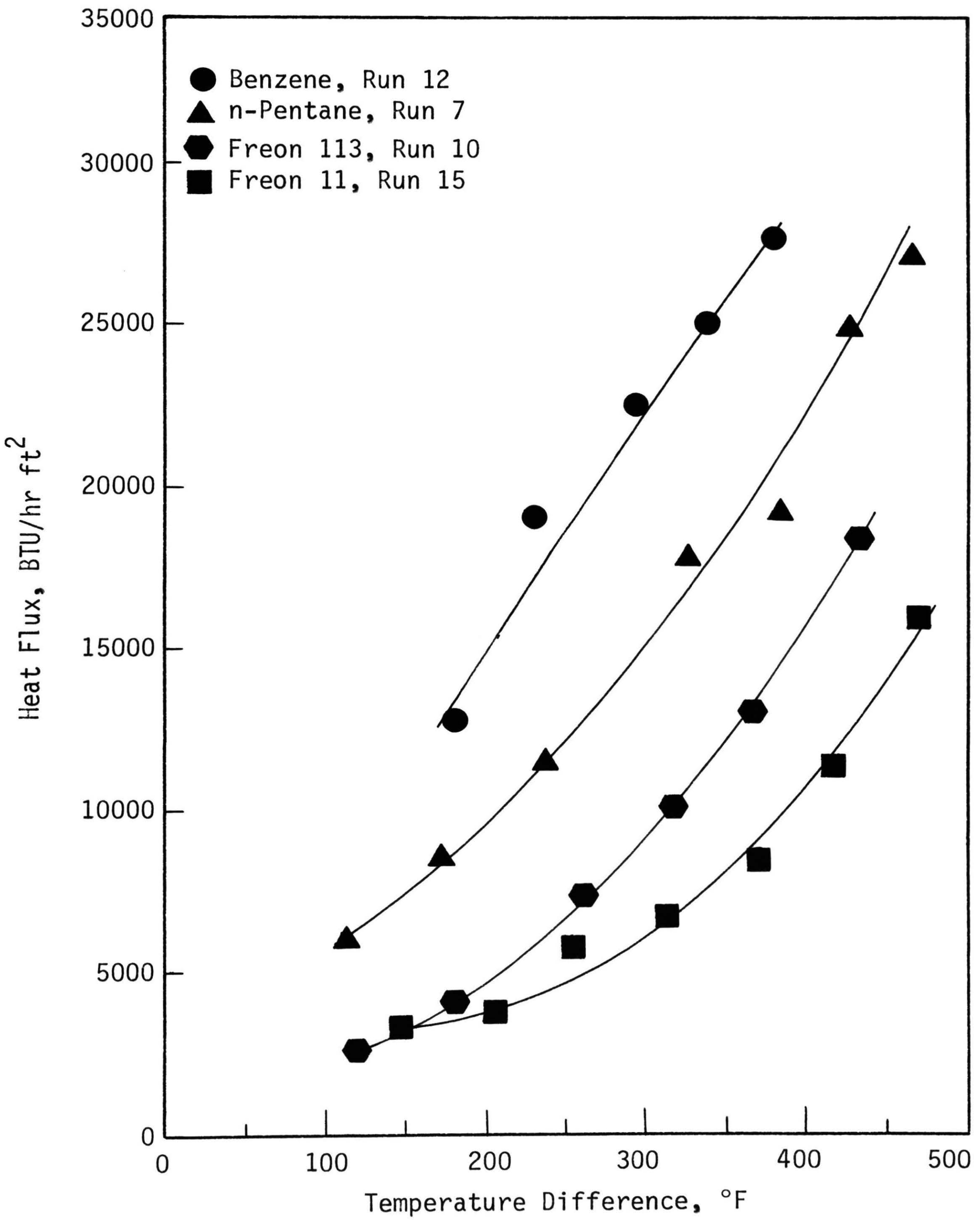


Figure 11: Film Boiling From a 2.045 Inch Diameter Surface

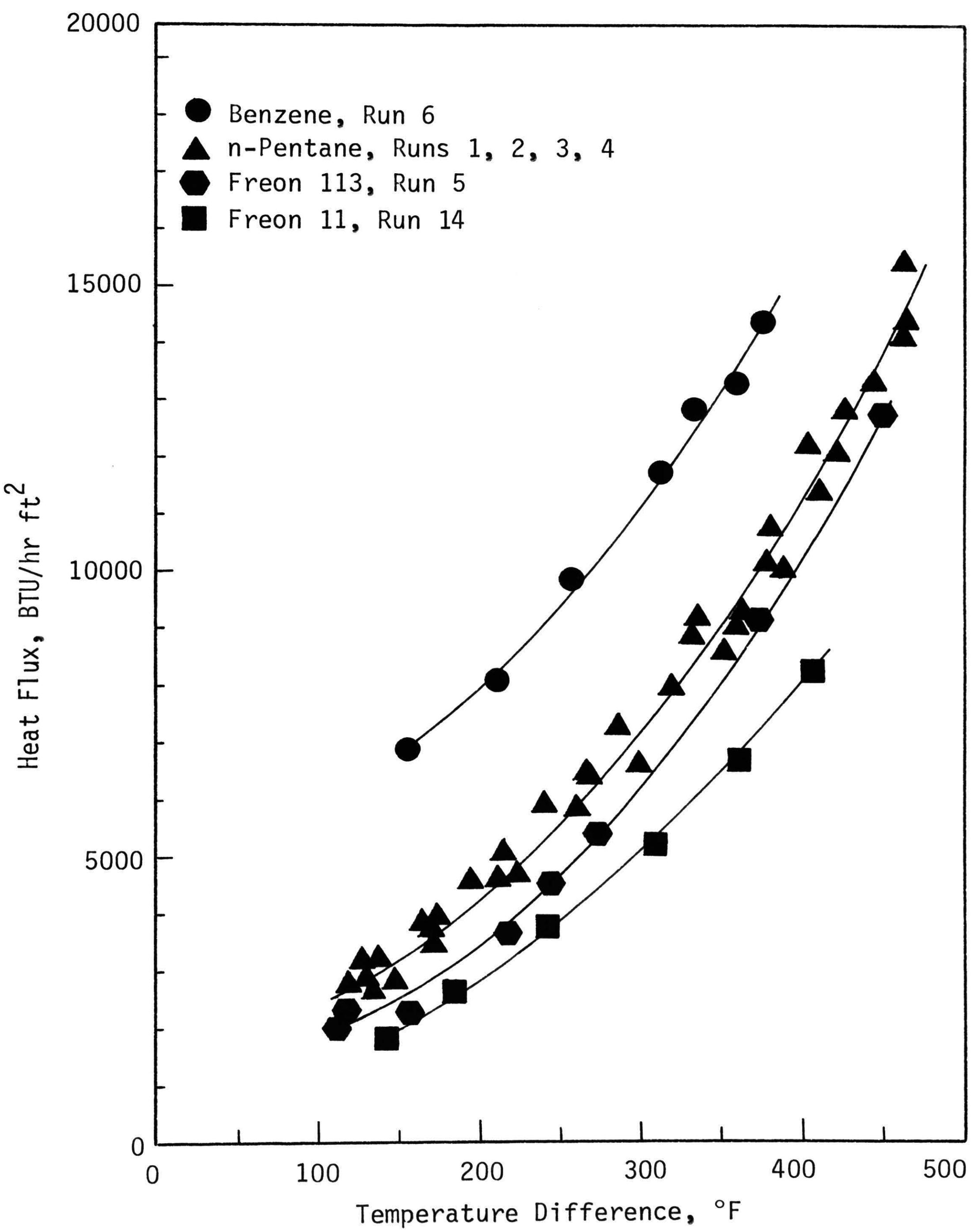


Figure 12: Film Boiling From a 3.025 Inch Diameter Surface

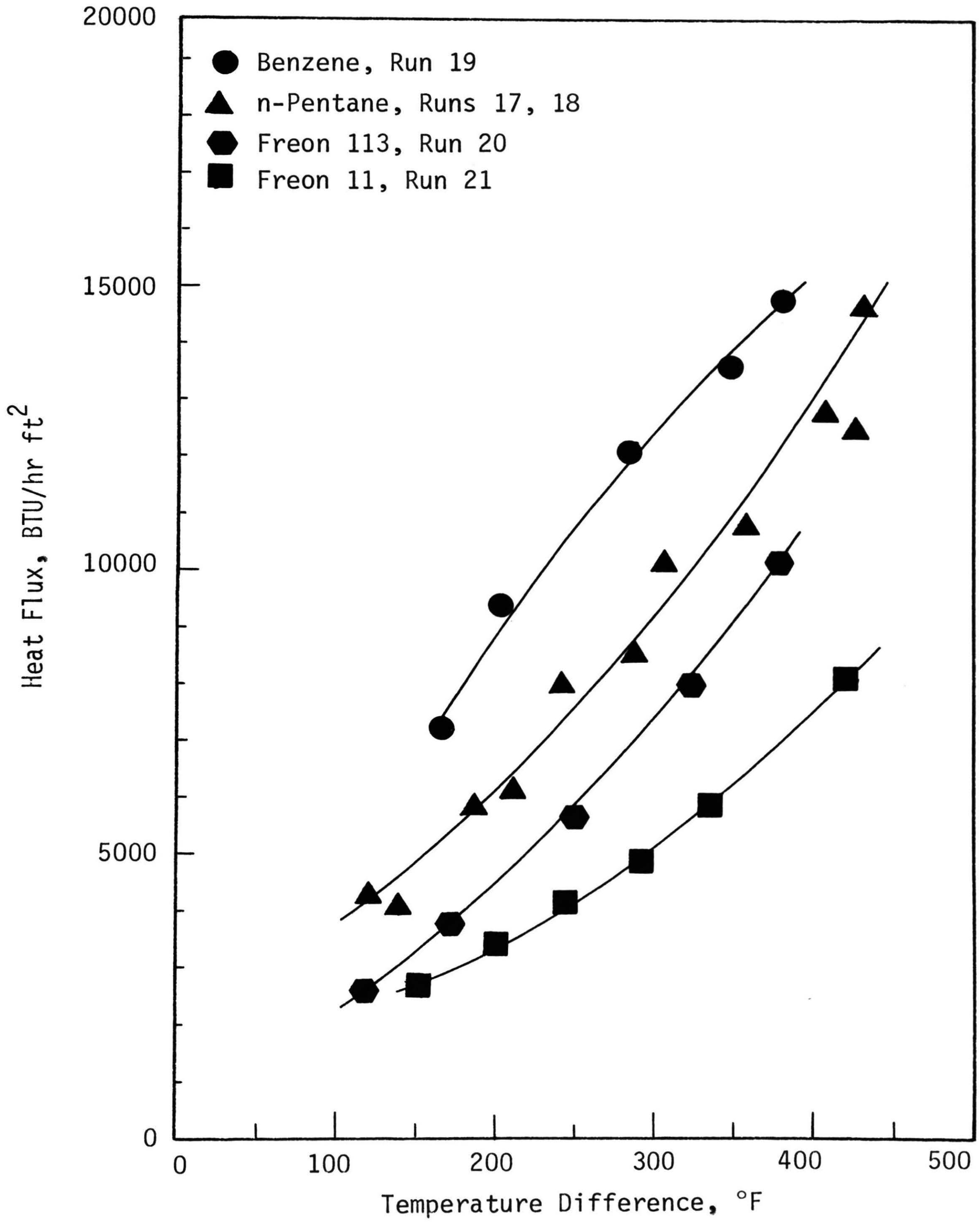


Figure 13: Film Boiling From a 3.630 Inch Diameter Surface

The uncertainty in the heat flux, based on heat losses measurements, is approximately fifteen percent.

It should be noted that the position of the data from the 0.4375 inch diameter surface, Figure 9, may not be reliable because of the high heat losses for this surface. The heat losses for this surface were on the order of ninety-five percent of the power input, and since the heat losses were determined experimentally, the uncertainty connected with this data is approximately 40,000 BTU/hour square foot for n-pentane at a temperature difference of four hundred degrees Fahrenheit.

VI. DISCUSSION

When the heat flux is plotted as a function of surface diameter at constant temperature difference, as in Figures 14 through 17, there appears to be a minimum level for heat flux existing for a surface of approximately three inches in diameter. Due to experimental uncertainty, it is not possible to say if this effect is actually a plateau, or if it is a slight dip before the heat flux levels off with respect to surface diameter. The area of the surface where this dip occurs is equal to the area of a three inch long cylinder of approximately 0.76 inches in diameter. Several investigators (28) (30) (31) have reported that a minimum does appear to exist for this size cylindrical heater. The dip may also be due to the ability of the larger heater surface to support several more departing bubble channels. It would thus allow a higher heat flux than a surface with a slightly smaller diameter. When the trend of the data is compared with that of Kermode and Zemaitis (21) for film boiling of benzene from a copper surface with an area of forty square inches, and Kautzky and Westwater's (26) data for film boiling of Freon 113 and Holser and Westwater's (25) data for Freon 11, both of the latter from an aluminum plate with an area of sixty-four square inches, in Figures 14, 16, and 17, respectively, it appears that a plateau has been reached with respect to surface area, and that a surface with a diameter greater than 3.630 inches can be considered an infinite plate. Figures 14 and 17 show the same minimum for all fluids. Since these surfaces were not circular, a hydraulic diameter

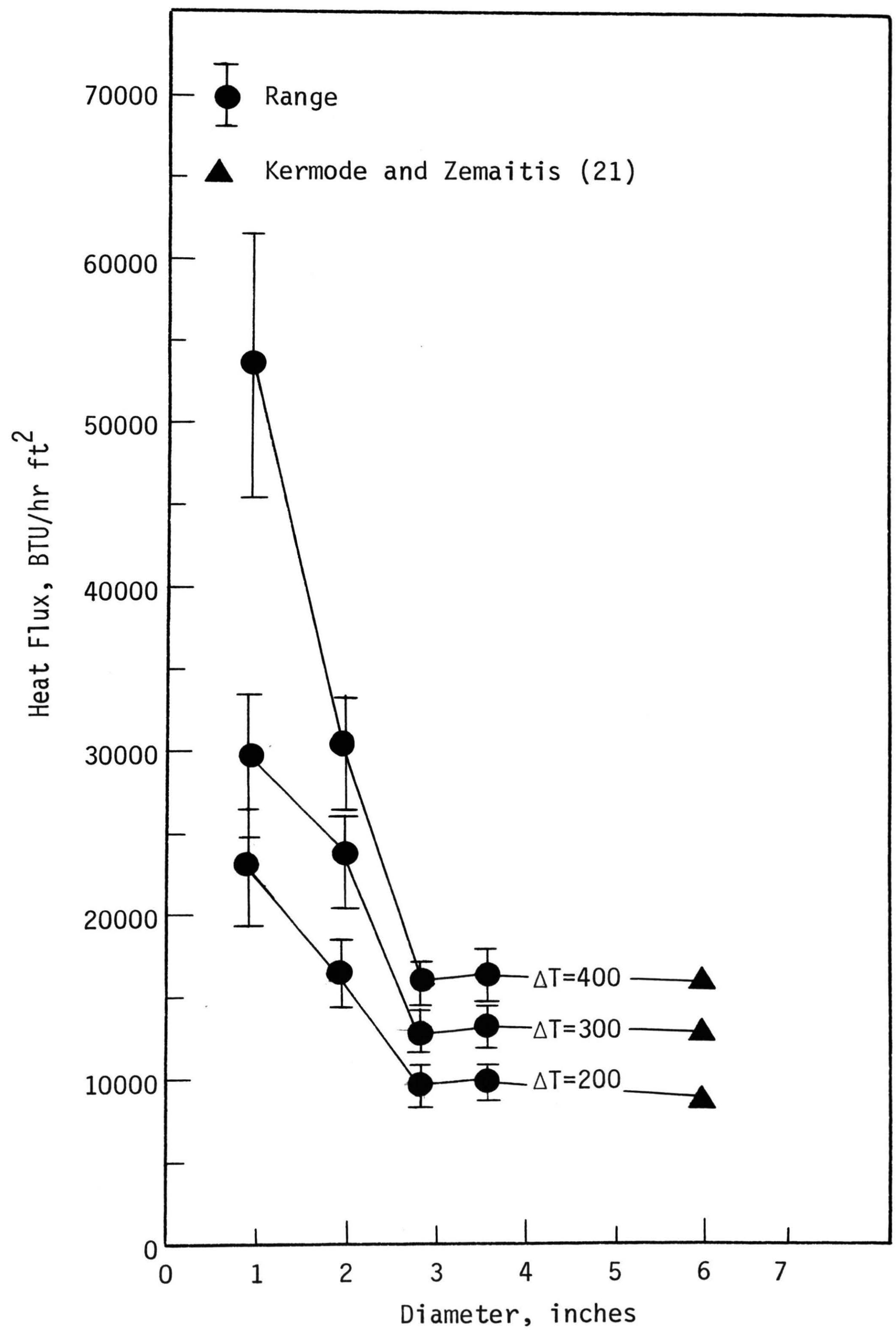


Figure 14: Diameter Effect on the Film Boiling of Benzene

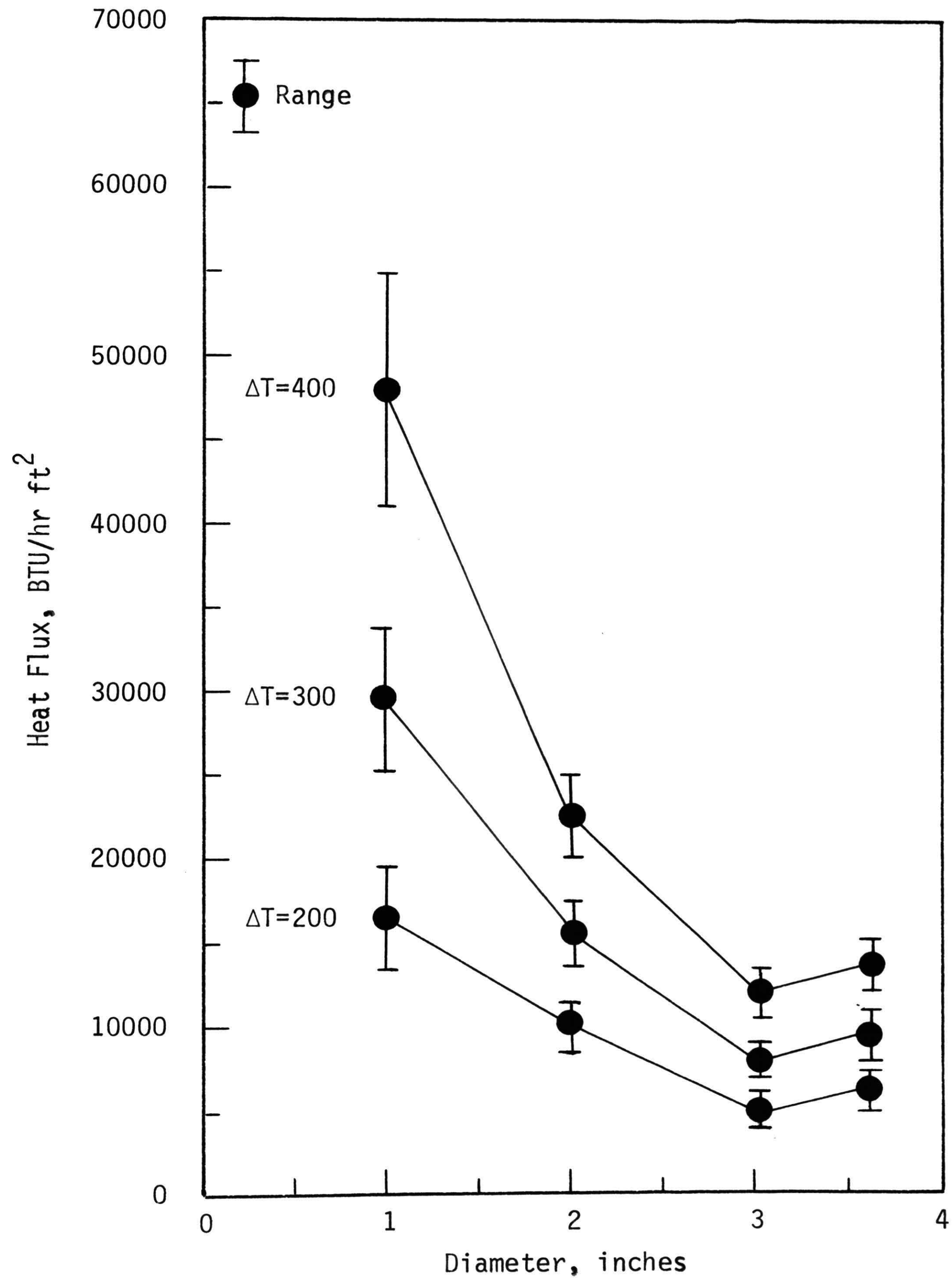


Figure 15: Diameter Effect on the Film Boiling of n-Pentane

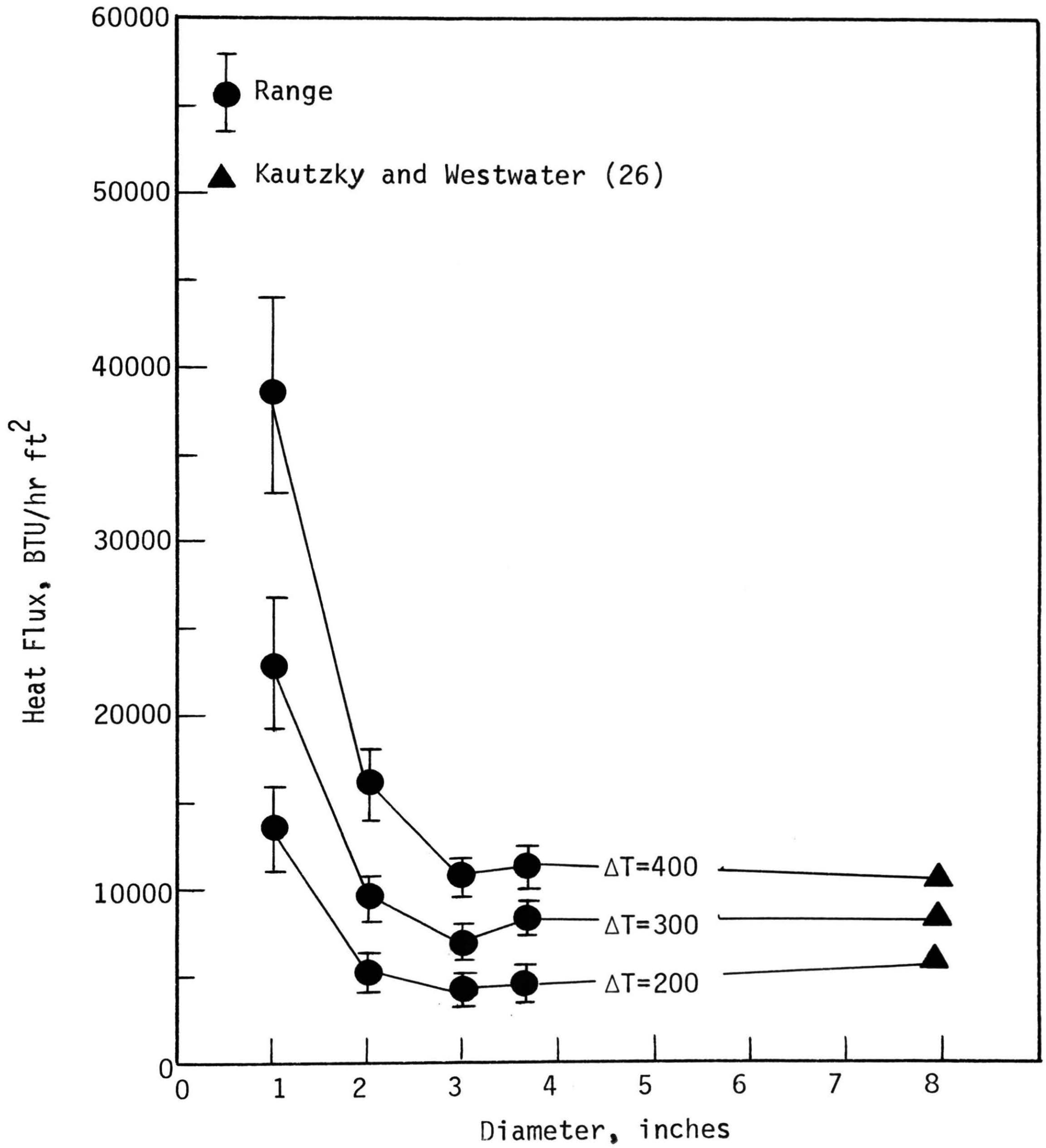


Figure 16: Diameter Effect on the Film Boiling of Freon 113

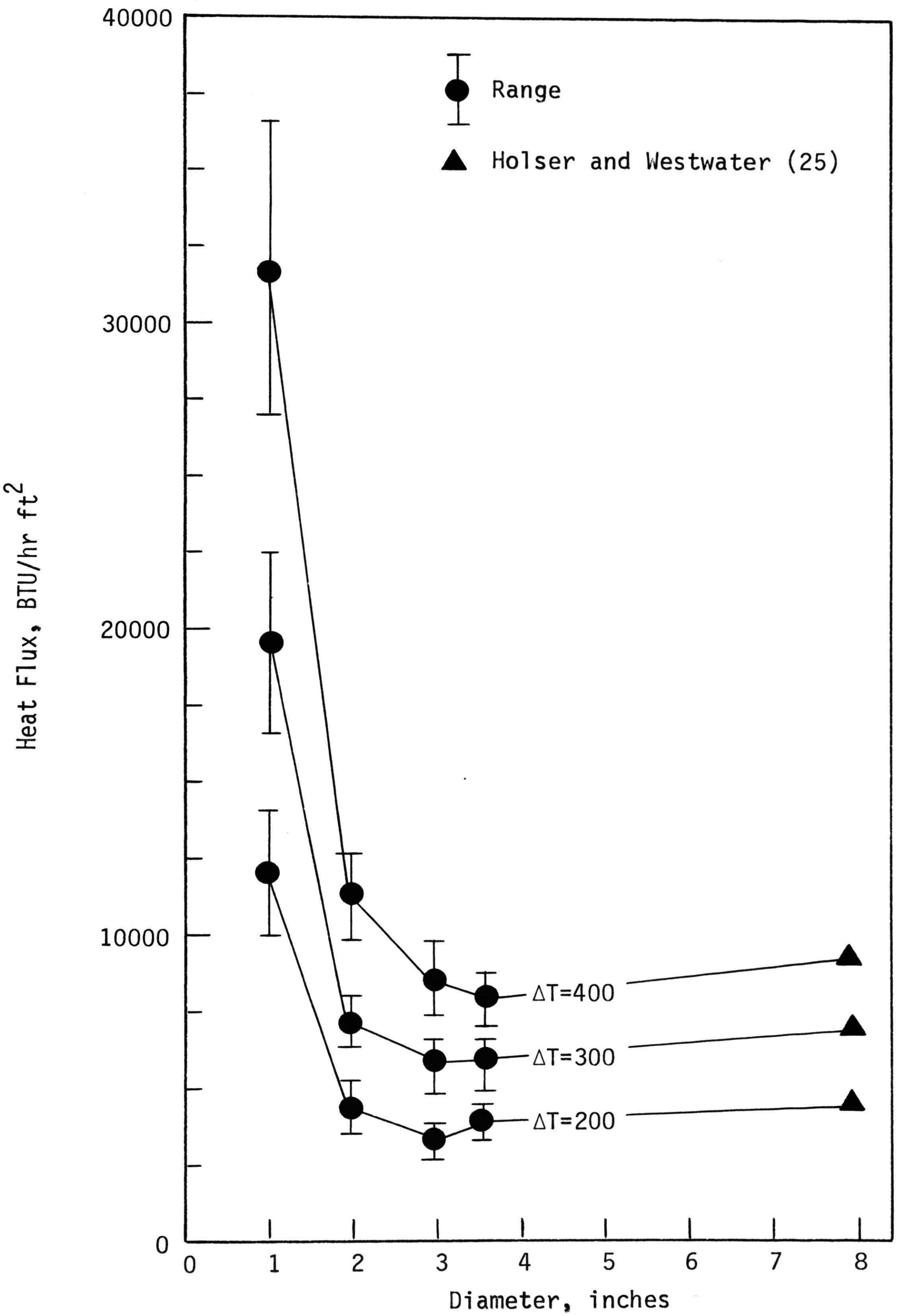


Figure 17: Diameter Effect on the Film Boiling of Freon 11

(four times the surface area divided by the perimeter length) was used when plotting the data.

Breen and Westwater (32) suggest that the minimum in the heat flux curve should occur at a diameter equal to the most dangerous wavelength, λ_d, for cylindrical heaters. Later work by Flanigan (30) and Kistler, Flanigan, and Park (31) indicates that the minimum is independent of λ_d. The λ_d of the test fluids varied from 0.456 to 0.698 inches, but the minimum point in the heat flux curve with respect to diameter is apparently a constant and not a function of the test fluid.

The range indicated in Figures 14 through 17 is based on the ten percent experimental uncertainty for these experiments. However, since the losses were determined experimentally, this limit was extended slightly. Based on the least squares curve fits used for the heat losses, the energy inputs could be approximately four percent in error. These errors mean that the position of each curve could be off by an amount proportional to the energy input. This amount, while relatively unimportant for the larger surfaces, is much larger for the smaller surface. The experimental uncertainty plus the correction for the uncertainty of the power input is represented as the range in Figures 14 through 17.

The hydrodynamic effects of boiling through a small channel, as was done in this investigation, have not been well characterized. After two runs were made with n-pentane on the 0.4375 inch diameter surface, a cylindrical depression with a diameter of two inches was cut in the boiling flange. This depression reduced the L/D of the boiling channel by one-half, from 2.286 to 1.143. N-pentane was then boiled again; the results given in Figure 18 show that the effects of this change are within experimental error, so that L/D effects are not present or are minimal. This test was with the smallest heater, therefore, channel effects would be greatest for this surface. Added verification of the negligible effect of the small channel from the surface into the boiling chamber can be seen by the comparison of the data of Kermode and Zemaitis (21), Kautzky and Westwater (26), and Holser and Westwater (25) as done in Figures 12, 14, and 15, respectively. These investigators utilized heaters that had no channel between the surface and the boiling fluid.

The theoretical equations of Chang, Frederking, and Berenson are shown along with the data from the 3.630 inch diameter surface in Figures 19 through 22. Based on these figures, Berenson's equation holds for Freon 113, Freon 11 and n-pentane for the largest surface. Since Berenson's equation was developed for n-pentane and carbon tetrachloride, this was expected. However, for benzene, Berenson's equation predicts values approximately forty-five percent too low. Since the molecular structure of benzene is much different from the structure of the other test fluids, it appears that some factor, based on structure, needs to be considered. The correlations of Chang and Frederking predict values approximately two times as great as the data, except in the case of benzene, where the correlations are only ten percent in error. Berenson's equation will fit the data

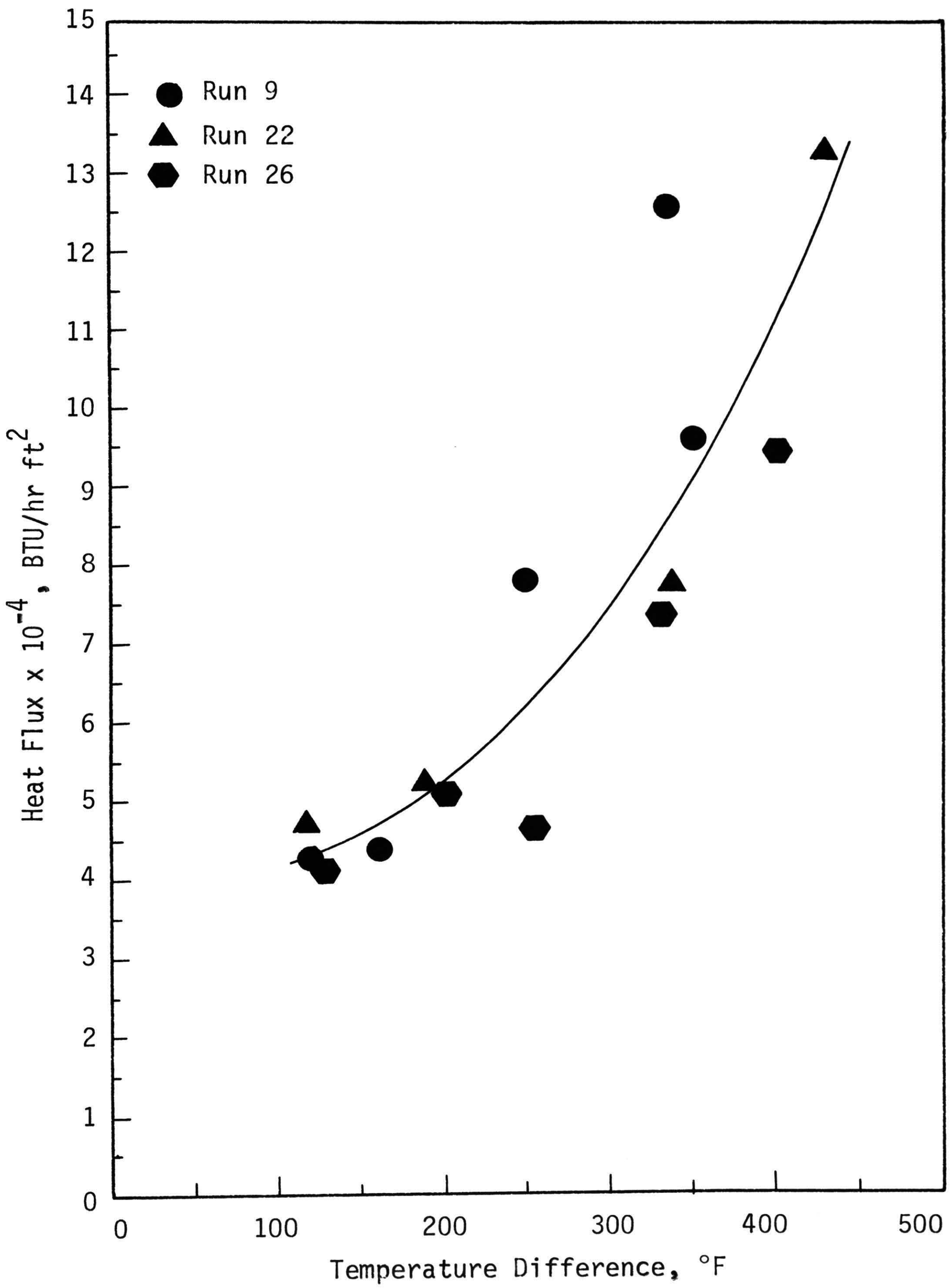


Figure 18: Film Boiling of n-Pentane on a 0.4375 Inch Diameter Surface

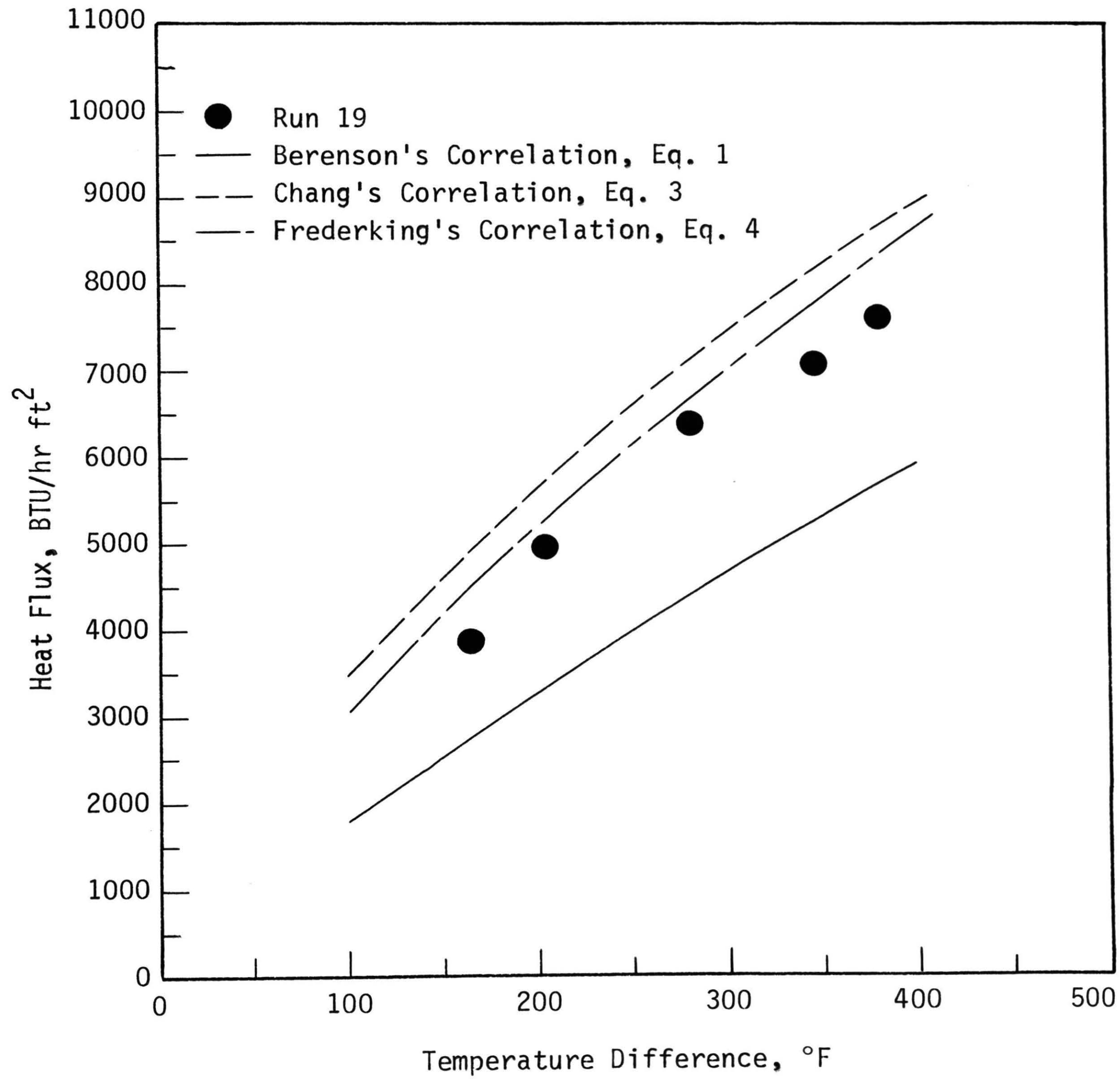


Figure 19: Film Boiling of Benzene from a 3.63 Inch Diameter Surface

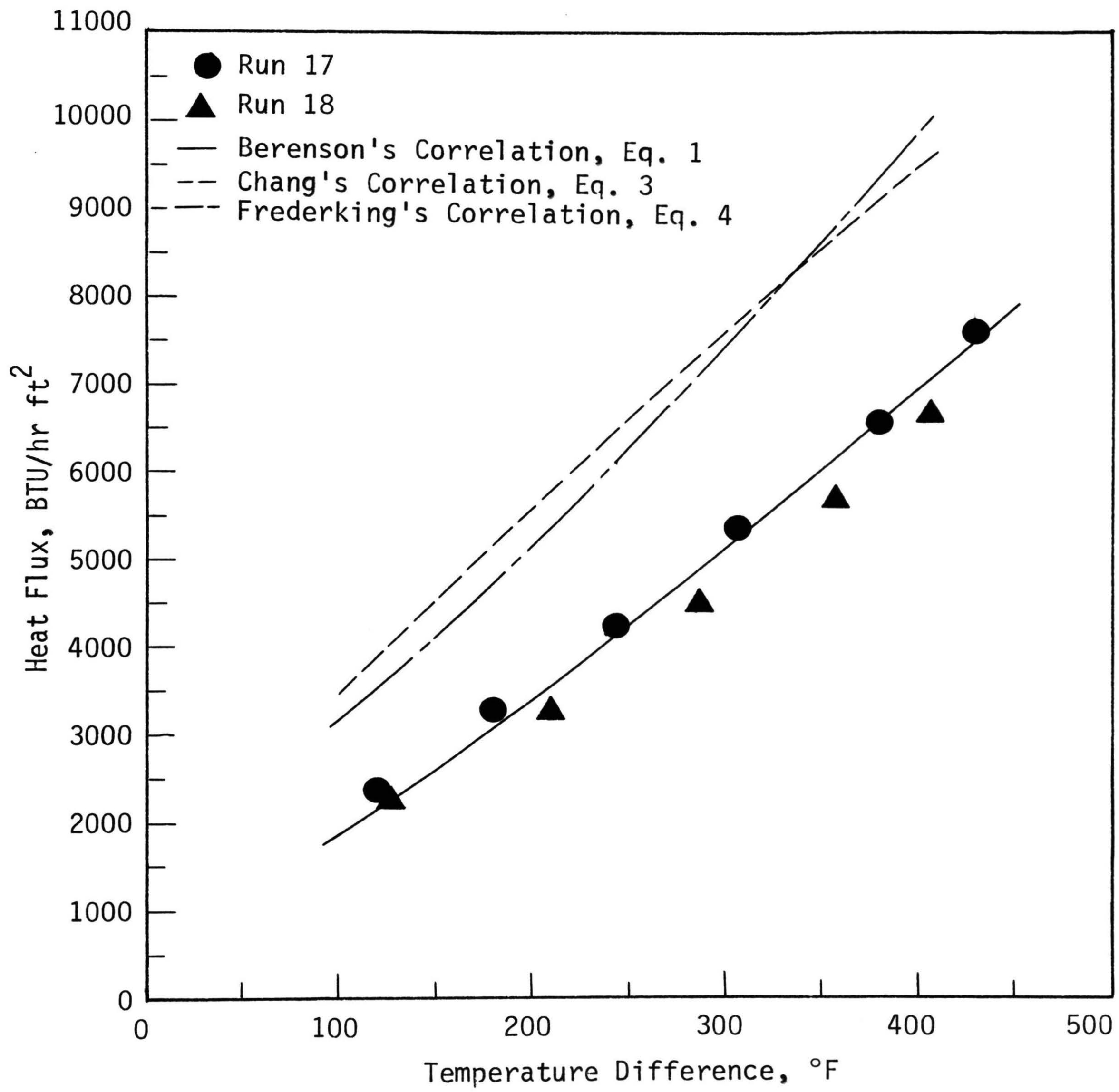


Figure 20: Film Boiling of n-Pentane from a 3.630 Inch Diameter Surface

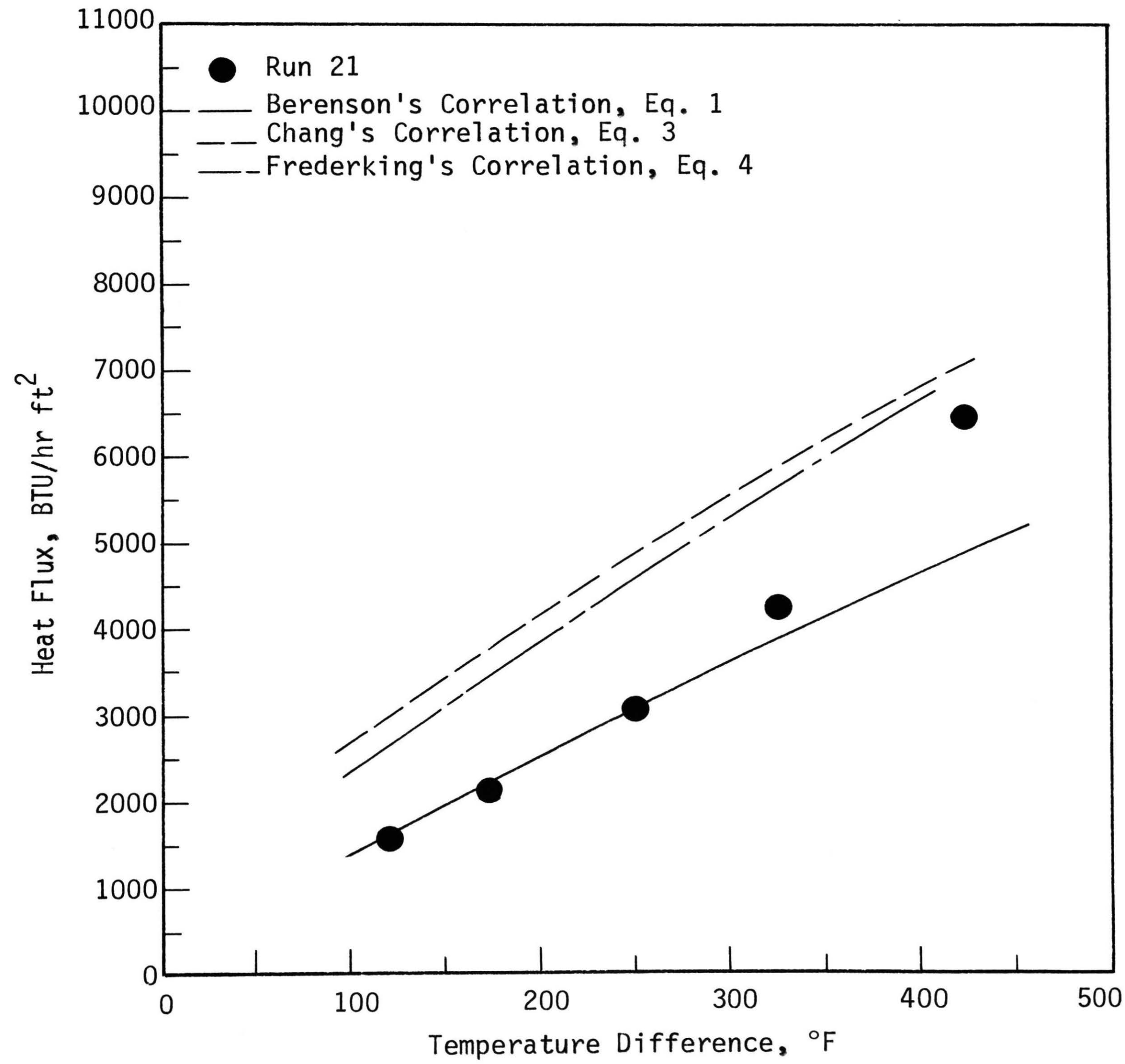


Figure 21: Film Boiling of Freon 113 From a 3.630 Inch Diameter Surface

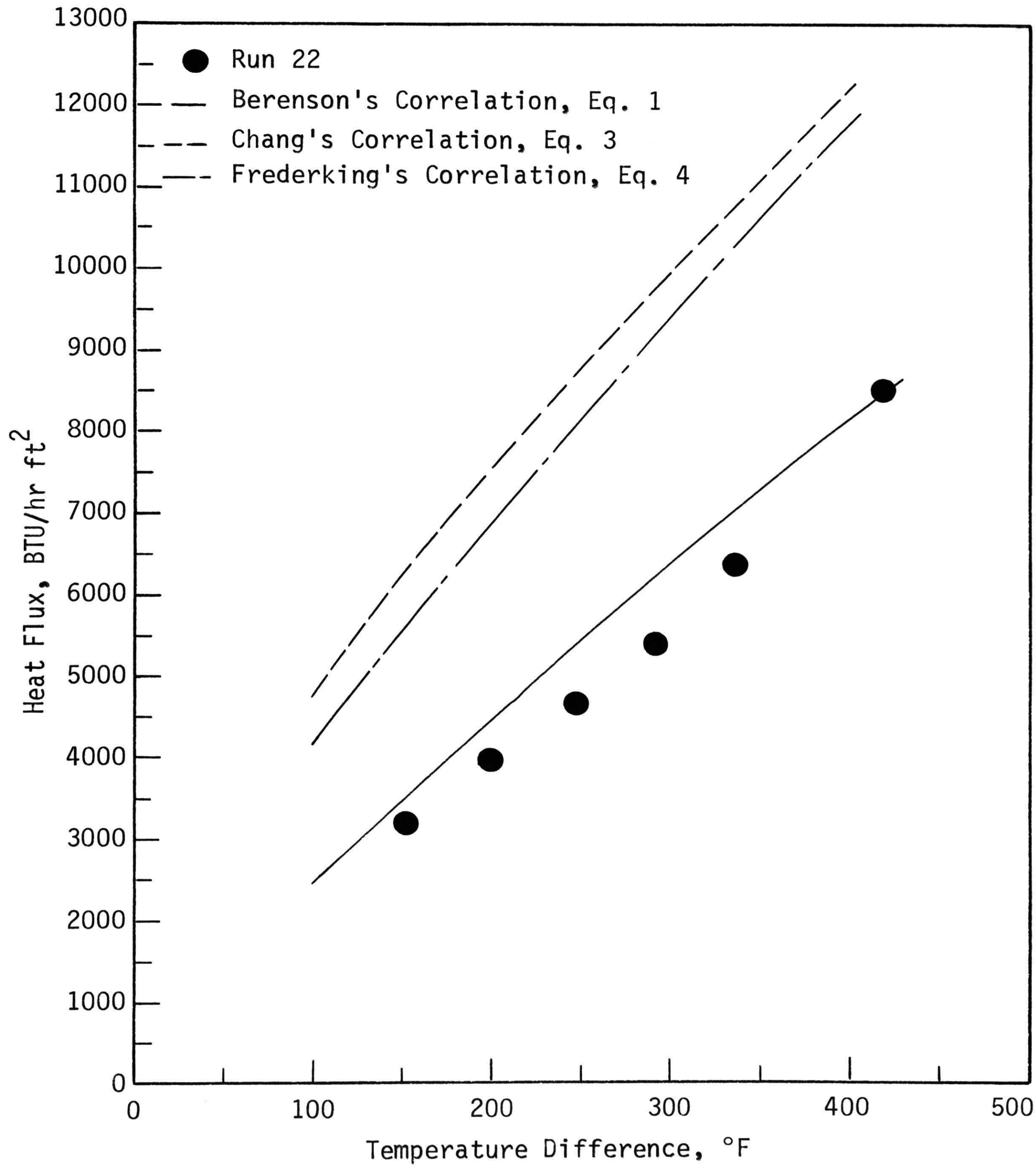


Figure 22: Film Boiling of Freon 11 from a 3.630 Inch Diameter Surface

for benzene if a constant factor of 1.376 is introduced into his equation. However, there is no justification for this based on any simple steric factor. The physical property data for evaluation of these correlations was taken from references (33) through (39).

Berenson's equation has no factor to account for the effect of diameter, therefore, it will not predict film boiling from small surfaces. Care should be exercised when using Berenson's equation to be sure that heat transfer surface diameter is greater than 3.630 inches where this equation is valid.

Based on the behavior of the heat flux with diameter, shown in Figures 14 through 17, a second degree polynomial was fitted to the data using the method of least squares. The resulting equation is:

$$a(d) = 6.113 - 3.472\ d + .5685\ d^2 \qquad .....5$$

where d is the surface diameter in inches. This equation is compared with the data for all fluids in Figures 23 through 26. This equation is used as a multiplier with Berenson's equation, so that the resulting correlation is:

$$h = a(d)\ \{0.425\ \{\frac{K^3 \Delta h\ \rho g(\rho_e - \rho)}{\mu \Delta T \sqrt{\frac{g_c \sigma}{g(\rho_e - \rho)}}}\}^{\frac{1}{4}}_{f}\} \qquad .....6$$

This equation fits the data of Freon 11, Freon 113, and n-pentane with an average deviation of 15.7 percent. The maximum deviation observed is 70.1 percent and 75.4 percent of the data deviate less than the average of 15.7 percent. This same equation will fit

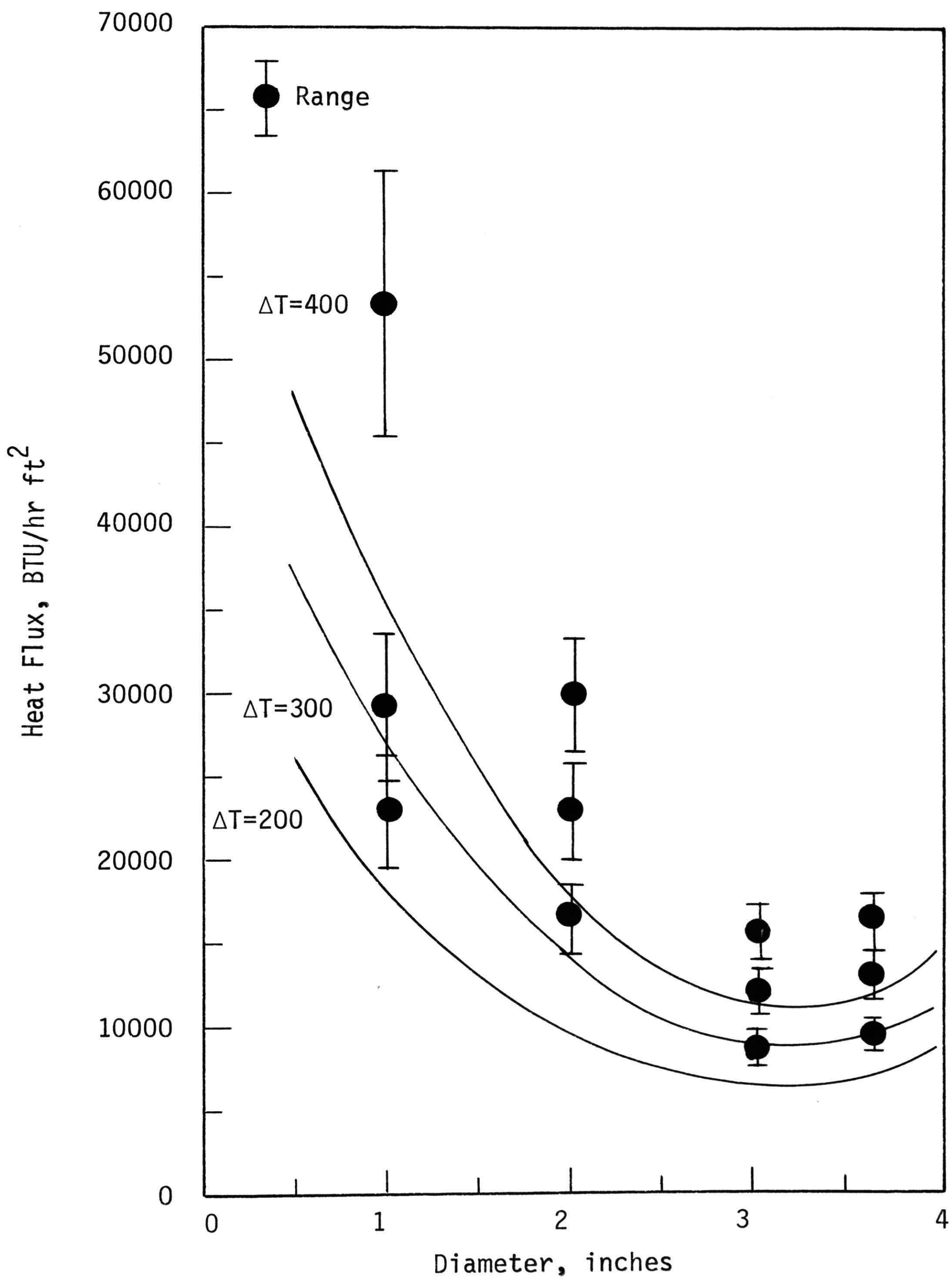


Figure 23: Diameter Correlation for Benzene

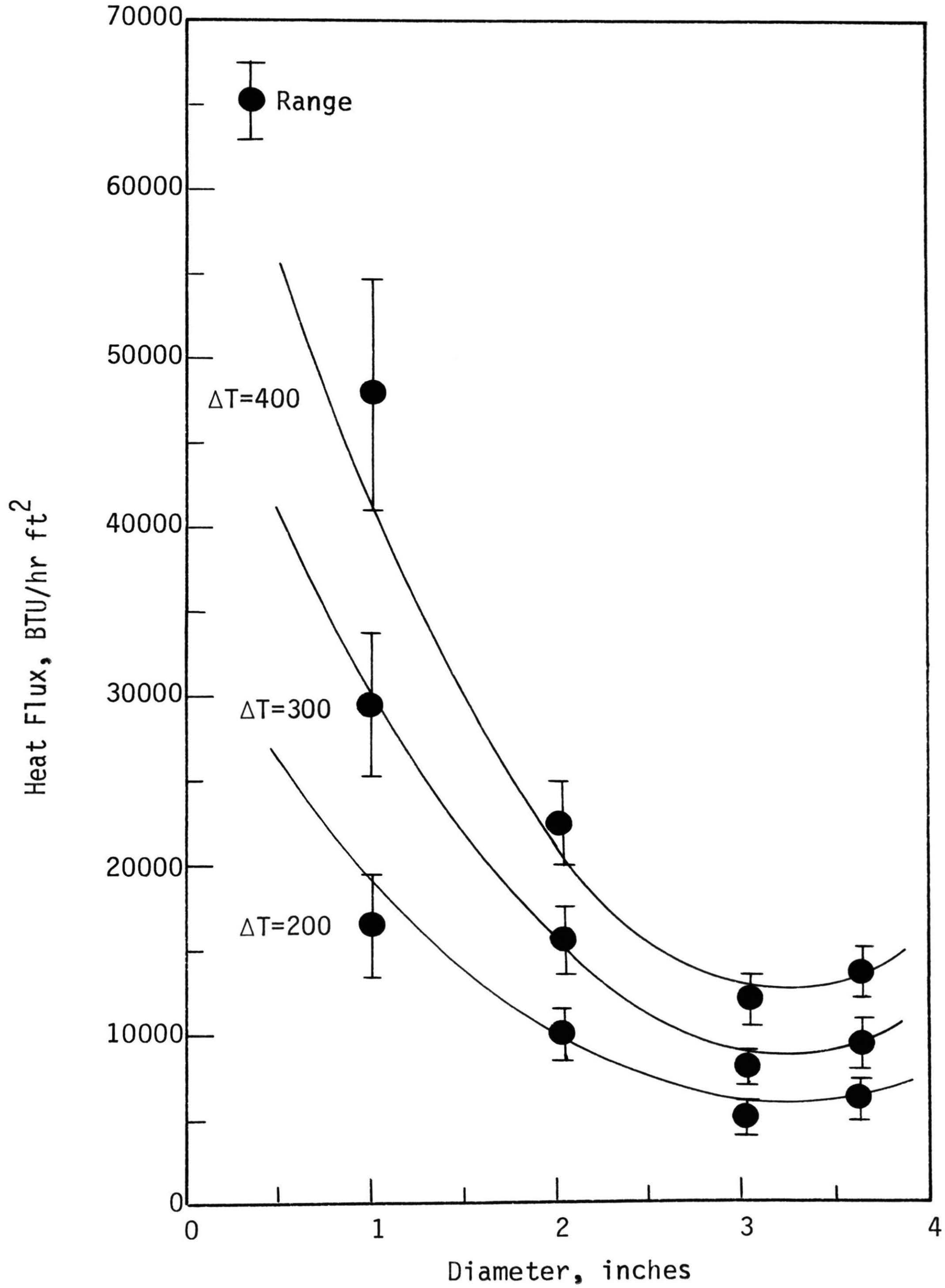


Figure 24: Diameter Correlation for n-Pentane

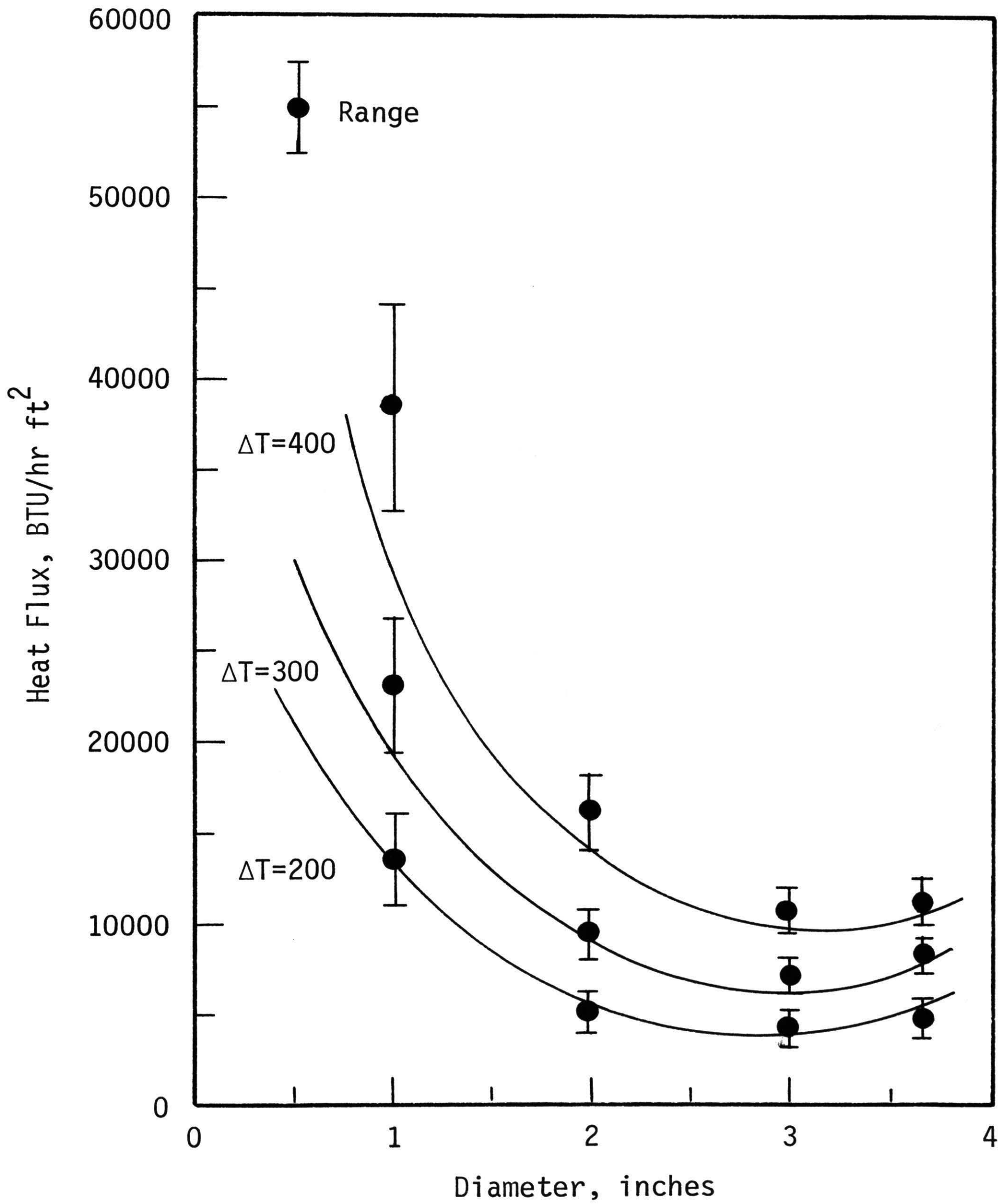


Figure 25: Diameter Correlation for Freon 113

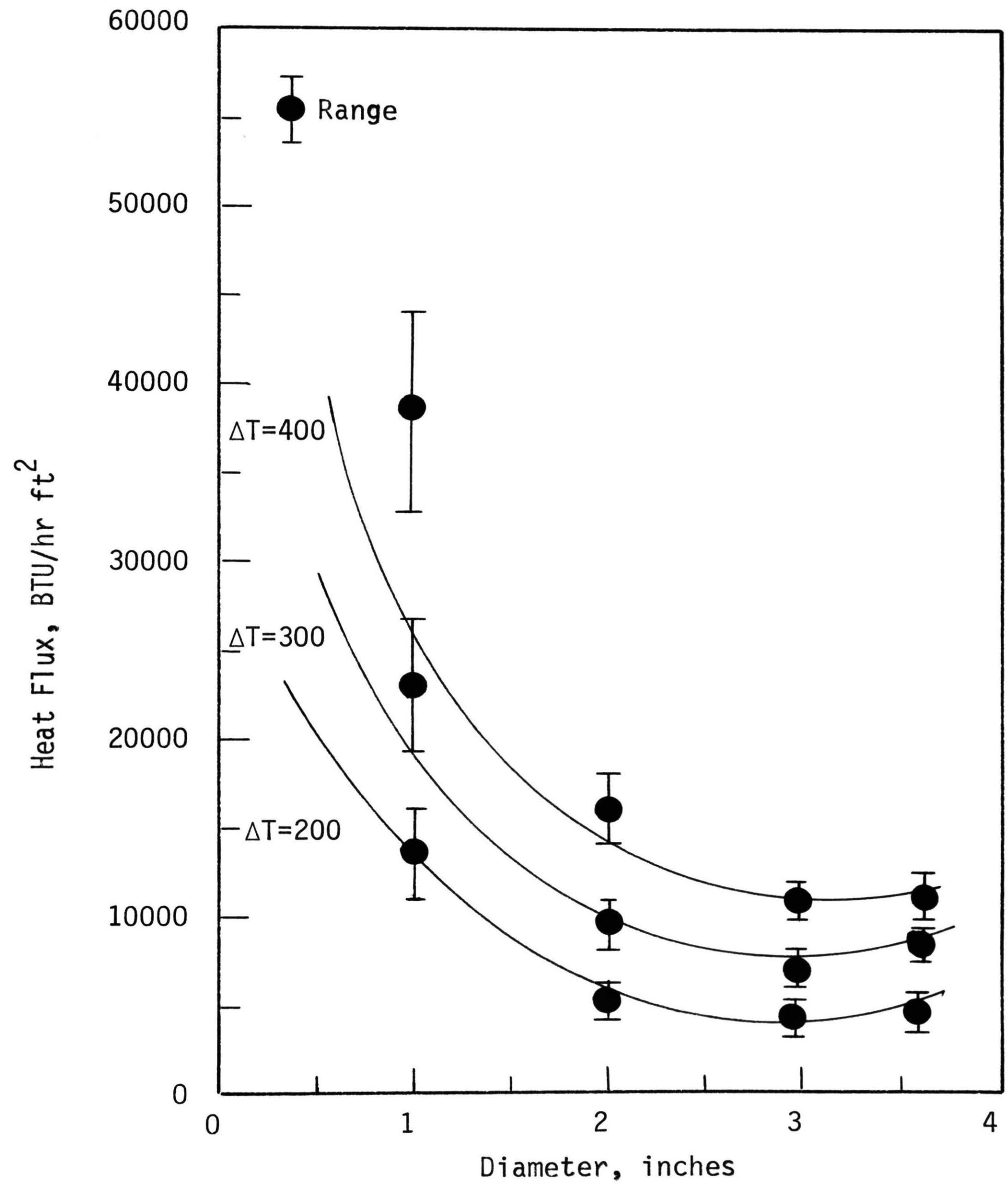


Figure 26: Diameter Correlation for Freon 11

benzene with approximately the same deviation if the constant factor of 1.376 is introduced.

This equation was developed for surfaces smaller than 3.630 inches in diameter for film boiling at atmospheric pressure. For surfaces larger than this, this multipling factor should not be used. This equation probably will not hold for pressures greater than atmospheric, or for fluids other than Freon 11, Freon 113, benzene, and n-pentane.

VII. CONCLUSIONS

1. For flat, horizontal surfaces with diameters of 3.025 inches or less, decreasing the surface diameter will result in an increase in the heat flux.

2. A minimum in the heat flux occurs at a surface diameter of approximately 3.025 inches. However, this minimum is not equal to the most dangerous wavelength, λ_d.

3. A film boiling correlation must have a diameter term to predict the film boiling behavior of small surfaces.

4. A flat plate with a diameter of 3.630 inches or greater may be considered an infinite plate in film boiling of benzene, Freon 11, Freon 113, and n-pentane.

5. Berenson's equation adequately predicts the film boiling behavior of Freon 113, Freon 11 and n-pentane from infinite flat plates at atmospheric pressure.

NOMENCLATURE

A = Heat transfer area, ft^2

ΔH = Latent heat of vaporization, BTU/(lb)

K = Thermal conductivity, BTU/(hr)(ft^2)(°F)

ρ = Density, $lb_m/(ft^3)$

μ = Viscosity, $lb_f/(ft)(hr)$

ΔT = Surface temperature - saturation temperature, (°F)

g_c = Gravitational constant, $lb_n/(lb_f)$ ft/(hr^2)

g = Acceleration of gravity, ft/(hr^2)

h = Heat transfer coefficient, BTU/(hr)(ft^2)(°F)

C_p = Specific heat, BTU/(lb)(°F)

d = Surface diameter, in.

a = Diameter relationship, unitless

Greek Symbols

λ_d = Most dangerous wavelength, ft

λ = Latent heat of vaporization, BTU/lb

λ' = Corrected latent heat of vaporization = $\lambda(1+0.5C_p\Delta T)$

Subscripts

f = Film

e = Liquid

BIBLIOGRAPHY

1. Nukiyama, S., "Maximum and Minimum Values of Heat Q Transmitted from Metal to Boiling Water Under Atmospheric Pressure," Int. J. of Heat and Mass Transfer, 9, p. 1419, 1966. (Translated from J. Japan Soc. Mech. Engr., 37, p. 367, 1934).

2. Jacob, M., Heat Transfer, I, John Wiley and Sons, Inc., New York, 1949.

3. McAdams, W. H., Heat Transmission, Third ed., McGraw-Hill Book Co., Inc., New York, 1954.

4. Rohsenow. W. M. (ed.), Developments in Heat Transfer, Mass. Inst. of Tech. Press, pp. 169-260, 1964.

5. Leppert, G. and C. C. Pitts, "Boiling," Advances in Heat Transfer, I, T. F. Irvin, Jr., and J. P. Hartnett (eds.), Academic Press, New York, 1964.

6. Colver, P. C. and L. D. Clements, "Natural Convection Film Boiling Heat Transfer," Ind. and Eng. Chem., 64, pp. 26-46, 1970.

7. Leppert, G., "Boiling Heat Transfer," Am. Soc. Nav. Eng. J., 73, pp. 331-340, 1961.

8. Brentari, E. G. and R. V. Smith, "Nucleate and Film Pool Boiling Design Equations for O_2, N_2, H_2, and He," Int. Adv. in Cryogenic Eng., 10, p. 324, 1964.

9. Jordan, D. P., "Film and Transition Boiling," Advances in Heat Transfer, V, T. F. Irvin, Jr., and J. P. Hartnett (eds.), Academic Press, New York, 1968.

10. Taylor, "The Instability of Liquid Surfaces when Accelerated in a Direction Perpendicular to their Plane, I," Proceedings, Royal Society of London, Series A, p. 192, 1950.

11. Berenson, P. J., "Experiments on Pool Boiling Transfer," Int. J. of Heat and Mass Transfer, 5, p. 985-999, 1962.

12. Zuber, N., "Hydrodynamic Aspects of Boiling Heat Transfer," AEUC-4439, 1959.

13. Zuber, N., "On the Stability of Boiling Heat Transfer," Trans. A.S.M.E., 80, 1958.

14. Lienhard and Wong, A.S.M.E. Paper 60-WA-147, 1960.

15. Berenson, P. J., "Film Boiling Heat Transfer From a Horizontal Surface," J. of Heat Transfer, 83, 1961.

16. Bromley, L. A., "Heat Transfer in Stable Film Boiling," CEP, 45, pp. 221-227, 1950.

17. Ruckenstein, E., "Film Boiling on a Horizontal Surface," Int. J. of Heat and Mass Transfer, 10, pp. 911-919, 1967.

18. Ragsdale, K. M., "Film Boiling from Horizontal Surfaces with Low Thermal Capacity," M.S. Thesis, University of Missouri-Rolla, 1967.

19. Ragsdale, K. M. and H. L. Sauer, Jr., "Film Boiling Heat Transfer to Refrigerant-11 from Flat Surfaces," ASHRAE Transactions, 75, pp. 126-132, 1969.

20. Hamill, T. D. and K. T. Baumeister, "Film Boiling Heat Transfer from a Horizontal Surface as an Optimal Boundary Valve Process," Proceedings, 3rd International Heat Transfer Conference, 4, pp. 59-65, 1966.

21. Kermode, M. L. and J. F. Zemaitis, Jr., "An Experimental Analysis of Pool Boiling from a Flat Horizontal Plate," Brit. Chem. Eng., 4, pp. 675-677, 1965.

22. Chang, Y. P., "Wave Theory of Heat Transfer in Film Boiling," J. of Heat Transfer, pp. 1-12, 1959.

23. Frederking, T. H. K., Y. C. Wu, and B. W. Clement, "Effects of Interfacial Instability on Film Boiling of Saturated Liquid Helium above a Horizontal Surface," A.I.Ch.E. J., 12, pp. 238-244, 1966.

24. Heath, C. A. and C. P. Costello, "Some Effects of Geometry, Orientation and Acceleration on Pool Boiling of Organic Fluids," J. of Engr. Ind., 82, p. 17-28, 1960.

25. Hosler, E. R. and J. W. Westwater, "Film Boiling on a Horizontal Plate," ARS J., 32, pp. 553-558, 1962.

26. Kautzky, D. E. and J. W. Westwater, "Film Boiling of a Mixture on a Horizontal Plate," Int. J. of Heat and Mass Transfer, 10, pp. 253-256, 1967.

27. Kesselring, R. C., P. H. Rosche, and S. G. Bankoff, "Transition and Film Boiling from a Horizontal Strips," A.I.Ch.E. J., 13, pp. 669-674, 1967.

28. Banchero, J. T., G. E. Barker, and R. H. Boll, "Stable Film Boiling of Liquid Oxygen Outside Single Horizontal Tubes and Wires," CEP Symposium Series, 17, pp. 21-30, 1955.

29. Kistler, R. S., D. V. Porchey, and E. L. Park, Jr., "Film Boiling of Liquid Nitrogen from Flat Plates over a Wide Range of Pressures," IIR Paper 2.22, 1971.

30. Flanigan, V. J., "A Study of Film Boiling of Liquid Nitrogen and Liquid Argon over a Wide Range of Pressures with Cylindrical Heaters," Ph.D. Thesis, University of Missouri-Rolla, 1967.

31. Kistler, R. S., V. J. Flanigan and E. L. Park, Jr., "Film Boiling of Liquid Nitrogen from Cylindrical Heaters," Presented, Missouri Academy of Science, 1971.

32. Breen, B. P. and J. W. Westwater, "Effect of Diameter of Horizontal Tubes on Film Boiling Heat Transfer," CEP, 15, p. 67, 1962.

33. Chemical Engineers Handbook, R. H. Perry, C. H. Chilton, and S. D. Kirkpatrick (eds.), McGraw-Hill Book Co., Inc., New York, 1963.

34. Bromley, L. A. and C. R. Wilke, "Viscosity Behavior of Casses," Ind. and Eng. Chem., 43, pp. 1621-1648, 1951.

35. Francis, A. W., "Pressure-Temperature-Liquid Density Relations of Pure Hydrocarbons," Ind. and Eng. Chem., 49, pp. 1779-80, 1957.

36. "Freon Fluorocarbon Properties and Applications," E. I. DuPont De Nemours and Co., Inc., 1969.

37. Reid, R. C. and T. K. Sherwood, The Properties of Gasses and Liquids, McGraw-Hill Book Co., Inc., New York, 1966.

38. Kobe, K. A. and E. G. Long, "Thermochemistry for the Petrochemical Industry," Pet. Refiner., 28, pp. 113-116, 1949.

39. Pitzer, K. S., "Thermodynamics of Gaseous Paraffins," Ind. and Eng. Chem., 36, pp. 829-831, 1944.

VITA

Richard Stanley Kistler, son of Mr. and Mrs. Ted P. Kistler, was born in Channute, Kansas, on October 3, 1947.

He attended Parkview High School in Springfield, Missouri, and graduated in June 1965. He enrolled in Drury College in 1965, Southwest Missouri State Teachers College in 1966, and the University of Missouri-Rolla in 1967. On May 24, 1967, he was married to Miss Bonnie Lee Cooke. He completed the requirements for a Bachelor of Science in Chemical Engineering in January 1970. He enrolled in Graduate School at the University of Missouri-Rolla, and completed requirements for a Master of Science degree in Chemical Engineering in December 1970.

APPENDIX A

CALCULATED FILM BOILING DATA

The calculated film boiling is presented in the sequence in which it was taken. Pressure recorder charts and a copy of the original data are in the possession of Dr. E. L. Park, Jr., Chemical Engineering Department.

TABLE I-A

Run 1

Calculated Film Boiling Data for n-Pentane on a 3.025 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$
129.42	3634.7	28.09	2512.5	304.54	123.14
140.09	3677.3	26.25	2568.8	320.62	137.09
167.48	4263.5	25.46	2858.2	385.70	172.91
199.29	5039.1	25.28	3095.2	466.01	214.52
242.57	6351.1	26.18	4103.2	588.08	271.10
289.26	7742.9	26.77	4674.0	718.62	332.17
336.12	9343.9	27.80	5703.0	859.79	393.44
384.03	11258.7	29.32	7233.8	1018.00	456.09
428.66	13268.4	30.95	9398.2	1176.72	514.51
448.39	13748.2	30.66	8253.0	1226.42	540.27
413.09	11846.2	28.68	7197.8	1085.33	494.10
363.04	9502.7	26.18	5907.0	902.92	428.66
270.42	6811.8	25.19	4179.0	647.50	307.53

TABLE II-A

Run 2

Calculated Film Boiling Data for n-Pentane on a 3.025 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$
121.71	3085.7	25.35	2574.0	267.06	113.06
136.48	3120.2	22.86	2954.2	288.00	132.38
150.30	3359.6	22.35	3033.0	318.13	150.45
174.96	3915.1	22.38	3490.5	378.09	182.69
225.78	5174.5	22.92	4612.2	507.41	249.16
299.73	7070.7	23.59	6066.0	698.75	345.86
364.59	9771.4	26.80	7746.8	918.36	430.68
421.30	12499.7	29.67	9180.8	1128.69	504.84
468.22	14815.4	31.64	10937.2	1305.62	566.19
393.00	10490.0	26.69	8580.8	991.39	467.84
265.83	5556.7	20.90	5593.5	578.85	301.52

TABLE III-A

Run 3

Calculated Film Boiling Data for n-Pentane on a 3.025 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$
119.44	3267.1	27.35	2201.2	273.15	110.10
132.00	3360.4	25.46	2851.5	294.23	126.52
177.35	4422.8	24.94	4006.5	406.56	185.82
218.43	5574.4	24.52	4871.2	517.75	239.54
263.61	6935.4	26.31	6062.2	644.76	298.63
336.52	9665.2	28.72	8031.0	876.35	393.97
407.52	12689.1	31.14	10208.0	1120.12	486.82
465.83	15917.8	34.17	11560.5	1357.51	563.07
383.13	10585.4	27.63	10187.2	983.23	454.92
322.35	8419.8	26.12	8052.8	795.66	375.43
265.66	6313.4	23.77	6512.2	616.40	301.30
213.45	5063.7	23.72	6781.5	485.78	233.05

TABLE IV-A

Run 4

Calculated Film Boiling Data for n-Pentane on a 3.025 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$
119.15	2886.4	24.22	2526.7	253.77	109.72	144.99
173.59	4220.1	24.31	3759.0	391.53	180.91	253.92
219.64	5220.7	23.77	5154.0	501.68	241.12	362.98
263.77	6254.9	23.71	6203.2	611.01	298.84	453.99
322.23	8521.4	26.45	7567.5	800.58	375.28	587.05
384.19	11104.9	28.91	9268.5	1010.54	456.30	730.51
467.36	14584.6	31.21	11595.8	1292.92	565.07	1006.49
353.83	9044.4	25.56	7889.2	868.01	416.61	748.42
247.55	5317.9	21.48	5032.5	543.03	277.62	493.69

TABLE V-A

Run 5

Calculated Film Boiling Data for Freon 113 on a 3.025 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
112.00	2434.9	21.74	2476.5	277.25	155.72	212.01	34.64
160.08	2665.1	16.65	2802.8	342.07	209.06	239.51	39.72
217.81	4198.0	19.27	3561.0	482.61	273.09	320.83	47.55
271.35	5857.2	21.58	4581.0	624.81	332.48	445.66	54.46
370.17	9629.0	26.01	6578.2	922.69	442.11	685.41	67.45
448.35	13234.0	29.52	8177.2	1189.33	528.84	883.09	77.40
244.03	5014.0	20.55	3789.8	552.43	302.19	511.50	42.91

TABLE VI-A

Run 6

Calculated Film Boiling Data for Freon 113 on a 3.025 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
158.04	7338.1	46.43	4389.8	605.39	239.16	332.82	84.42
210.70	8548.5	40.57	5575.5	724.10	297.45	459.33	88.28
258.51	10308.7	39.88	6651.8	864.88	350.39	570.82	92.24
311.50	12198.7	39.16	7884.0	1017.89	409.06	718.25	97.28
361.01	13784.6	38.18	8935.5	1151.86	463.88	803.74	102.30
377.26	14832.7	39.32	9532.5	1222.16	481.87	929.97	99.88
335.48	13257.4	39.52	8846.2	1097.27	435.61	902.94	91.84

TABLE VII-A

Run 7

Calculated Film Boiling Data for n-Pentane on a 2.045 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
115.39	6451.4	55.91	2715.8	251.95	104.80	116.61	28.01
174.70	8956.4	51.27	4082.2	386.64	182.35	251.12	33.32
239.69	12017.4	50.14	5764.5	541.45	267.34	359.80	42.15
329.57	18184.4	55.18	7825.5	799.66	384.89	527.86	55.34
428.66	25444.2	59.36	10411.5	1094.83	514.46	772.96	68.28
469.05	27599.8	58.84	11552.2	1196.82	567.28	898.14	70.58
386.74	19610.64	50.71	9732.0	906.96	459.65	754.08	55.57
319.32	15748.0	49.32	7896.0	730.68	371.48	607.93	45.25

TABLE VIII-A

Run 8

Calculated Film Boiling Data for n-Pentane on a 1.016 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
140.62	13601.8	96.73	2146.5	214.36	137.79	87.96	31.95
183.71	14203.1	77.31	3246.0	274.10	194.14	163.61	34.54
244.05	21703.2	88.93	4272.0	395.23	373.04	249.34	42.88
337.87	34357.5	101.69	6138.8	589.17	395.73	410.78	53.93
420.50	53660.7	127.61	7455.8	805.91	503.79	521.51	65.59
466.57	64471.0	138.18	8221.5	927.01	564.04	600.70	71.48
386.09	38064.8	98.59	6894.0	673.11	458.80	529.38	55.58

TABLE IX-A

Run 9

Calculated Film Boiling Data for n-Pentane on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
122.10	43348.6	355.02	1544.2	158.79	113.57	54.13	29.03
162.49	44578.9	274.35	1954.5	212.88	166.39	84.01	33.44
250.14	78994.2	315.80	2780.2	363.40	281.01	167.50	45.23
337.68	126030.0	373.22	3660.0	526.94	395.49	284.47	56.60
448.05	214283.3	478.25	5346.0	763.32	539.82	437.34	70.76
353.83	96407.0	272.46	4666.5	517.17	416.61	373.40	50.79

TABLE X-A

Run 10

Calculated Film Boiling Data for Freon 113 on a 2.045 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
120.74	3031.9	25.11	2194.5	234.57	165.41	56.32	42.06
180.81	4556.2	25.20	3092.2	335.97	232.05	165.08	47.63
263.53	7837.4	29.74	5201.2	502.58	323.82	305.40	55.22
318.35	10578.3	33.23	6844.5	629.92	384.63	304.47	62.15
369.58	13412.6	36.29	7935.8	747.40	441.46	491.62	69.13
435.78	18811.4	43.17	9567.0	943.97	514.90	629.45	78.00

TABLE XI-A

Run 11

Calculated Film Boiling Data for Freon 113 on a 1.016 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
116.90	9837.2	84.15	3084.8	216.54	161.15	79.03	38.28
180.85	12538.3	69.33	3380.2	302.69	232.09	126.71	44.85
272.13	19089.4	70.15	3610.5	440.83	333.36	250.18	52.96
345.13	28816.6	83.49	4520.2	576.58	414.34	364.51	60.83
382.72	34524.7	90.21	4819.5	650.41	456.03	419.86	64.60
433.52	45035.2	103.88	5574.0	765.94	512.39	497.97	70.39

TABLE XII-A

Run 12

Calculated Film Boiling Data for Benzene on a 2.045 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
181.88	13227.4	72.73	5509.5	567.25	265.54	256.24	89.28
232.24	19483.4	83.89	7383.8	765.71	321.31	430.84	94.66
294.96	22986.4	77.93	8511.8	915.05	390.74	549.37	101.66
340.87	25542.3	74.93	9032.2	1024.19	441.58	634.61	106.45
381.52	28128.8	73.73	9697.5	1128.19	486.59	711.02	110.76
325.74	23138.7	71.03	9186.0	952.61	424.83	701.30	92.98

TABLE XIII-A

Run 13

Calculated Film Boiling Data for Benzene on a 1.016 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
181.46	17031.2	93.86	2683.5	360.97	265.08	73.08	83.55
232.61	26733.3	114.93	3329.2	472.22	321.71	160.45	87.65
293.34	39203.3	133.64	4665.0	609.68	388.96	264.47	92.62
328.74	45623.8	138.78	5291.2	685.01	428.15	316.62	95.40
365.56	53763.0	174.09	6065.2	771.61	468.92	382.56	99.20
388.65	57769.1	148.64	6180.8	819.73	494.48	424.26	100.88

TABLE XIV-A

Run 14

Calculated Film Boiling Data for Freon 11 on a 3.025 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
144.49	2280.5	15.78	2339.2	249.10	135.28	185.31	20.82
185.45	3123.2	16.84	2966.2	346.97	191.09	273.77	27.31
243.72	4283.3	17.57	3815.2	484.26	270.48	361.68	36.54
311.27	5719.5	18.37	5085.0	647.99	362.53	489.26	47.28
361.55	7176.9	19.85	6156.0	789.23	431.03	567.37	55.22
408.32	8704.0	21.32	7266.8	929.18	494.77	687.18	62.66

TABLE XV-A

Run 15

Calculated Film Boiling Data for Freon 11 on a 2.045 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
147.49	3774.0	25.59	1871.2	225.44	139.36	148.75	21.30
205.53	4274.4	20.80	2507.2	315.95	218.45	241.50	30.50
257.14	6192.3	24.08	3414.8	430.01	288.77	294.21	38.65
315.77	7173.4	22.72	4065.8	532.28	368.67	335.24	47.97
370.97	8846.2	23.85	5149.5	645.66	443.88	475.99	56.60
419.85	11799.9	28.10	5753.2	779.63	510.48	532.35	64.52
471.58	16393.5	34.76	7531.5	954.90	580.97	792.01	72.46

TABLE XVI-A

Run 16

Calculated Film Boiling Data for Freon 11 on a 1.016 Inch Diameter Surface at 14.7 psia Pressure

ΔT ($^\circ F$)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ ^\circ F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
151.10	10673.1	70.64	1979.2	204.37	144.28	131.48	21.92
217.36	12609.0	58.01	2590.5	305.55	234.56	197.85	32.48
295.18	18818.9	63.75	3921.8	446.56	340.61	275.08	44.87
361.96	23858.6	65.92	4587.8	565.92	431.60	367.35	55.54
432.59	36708.9	84.86	5421.8	734.52	527.85	454.65	66.76
472.13	45254.0	95.85	6951.0	836.50	581.72	552.18	73.04

TABLE XVII-A

Run 17

Calculated Film Boiling Data for n-Pentane on a 3.630 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
123.36	4751.3	38.52	2569.8	456.69	115.22	364.23	29.28
188.73	6322.7	33.50	4013.8	655.10	200.70	522.71	37.45
244.00	8424.8	34.53	5320.9	878.46	272.98	675.39	44.41
308.75	10612.8	34.37	7211.7	1120.40	357.66	901.08	51.94
380.00	13123.9	34.54	9708.0	1394.03	450.83	800.46	60.04
432.06	15156.2	35.08	10678.6	1608.16	518.90	1305.01	64.64

TABLE XVIII-A

Run 18

Calculated Film Boiling Data for n-Pentane on a 3.630 Inch Diameter Surface at 14.7 psia Pressure

ΔT ($^{\circ}F$)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ {}^{\circ}F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
141.54	4566.9	32.27	2145.2	467.21	138.99	235.10	32.76
213.23	6570.4	30.81	2228.1	704.95	232.74	478.57	41.83
289.18	9007.6	31.15	4742.9	979.43	332.07	730.79	49.61
359.45	11317.7	31.49	7498.5	1237.36	432.96	942.18	56.96
408.11	13291.6	32.57	8659.2	1442.84	487.59	1053.10	61.00

TABLE XIX-A

Run 19

Calculated Film Boiling Data for Benzene on a 3.630 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
166.18	7723.2	46.47	1324.8	803.23	248.17	383.27	89.77
203.91	9871.2	48.41	1687.4	999.38	289.94	735.77	87.70
282.87	12606.1	44.57	2164.2	1283.35	377.36	997.93	112.57
345.87	14107.4	40.79	3443.2	1461.00	447.11	1022.81	104.10
381.04	15229.4	39.97	3717.0	1580.58	486.06	1185.79	107.02

TABLE XX-A

Run 20

Calculated Film Boiling Data for Freon 113 on a 3.630 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
119.65	3092.8	25.85	869.2	386.49	164.21	123.34	43.27
173.67	4208.3	24.23	1357.7	526.58	224.13	258.11	50.55
252.31	6071.3	24.06	1768.1	747.71	311.37	489.68	59.90
326.72	8514.7	26.06	2193.5	1005.86	393.92	671.68	67.95
426.78	12954.1	30.35	4541.4	1435.91	504.91	1186.55	78.71

TABLE XXI-A

Run 21

Calculated Film Boiling Data for Freon 11 on a 3.630 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
153.88	3147.3	20.45	1703.7	374.26	148.07	264.55	22.35
201.18	3895.3	19.36	2206.9	492.47	212.52	347.53	29.86
249.68	4623.9	18.52	2609.2	610.93	278.61	455.67	37.60
293.45	5379.4	18.33	2922.7	724.86	338.25	550.08	44.54
338.80	6307.2	18.62	3743.1	853.34	400.04	668.42	51.71
421.46	8512.8	20.20	5506.7	1124.49	512.68	837.90	64.74

TABLE XXII-A

Run 22

Calculated Film Boiling Data for n-Pentane on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
119.39	49652.0	415.88	6331.2	161.86	110.03	10.72	28.59
192.86	52766.1	273.59	7504.5	261.20	206.11	80.30	38.50
266.55	48990.8	183.80	9570.1	353.61	302.46	187.80	44.61
343.13	78490.8	228.75	10909.0	484.56	402.62	255.60	54.36
435.61	133304.0	306.02	13027.7	662.72	523.55	350.02	66.85

TABLE XXIII-A

Run 23

Calculated Film Boiling Data for Benzene on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
175.36	(1819.9)	(10.38)	4310.5	256.43	258.33	64.85	84.27
231.75	24768.8	106.88	6673.2	346.62	320.76	85.46	85.03
300.52	49094.2	163.36	5217.2	448.16	396.91	154.40	92.37
336.89	76550.8	227.24	9891.7	517.07	437.15	224.71	96.49
390.56	146491.8	375.08	11940.5	649.53	496.60	275.77	104.22

TABLE XXIV-A

Run 24

Calculated Film Boiling Data for Freon 113 on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
120.17	11422.3	95.05	8674.4	176.71	164.79	28.89	40.81
226.47	(12999.3)	(57.40)	7852.1	269.13	282.70	128.45	47.07
255.61	(10186.9)	(39.85)	8424.3	304.39	315.03	163.13	46.01
317.30	11904.4	37.52	9008.0	395.89	383.47	215.85	53.69
422.22	79488.2	188.26	10655.3	582.83	499.85	332.88	67.67

TABLE XXV-A

Run 25

Calculated Film Boiling Data for Freon 11 on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
156.61	(5485.1)	(35.02)	4179.1	146.07	151.79	30.92	21.27
206.91	6143.8	29.69	4857.7	226.74	220.23	99.40	30.75
275.32	(6046.7)	(21.96)	6220.9	307.23	313.54	174.75	41.64
315.27	(6718.6)	(21.3103)	7119.6	360.97	367.98	221.57	47.95
391.31	10265.7	26.23	9085.1	482.31	471.60	303.21	59.98
460.71	50574.1	109.78	14968.6	618.95	556.15	393.53	70.95

TABLE XXVI-A

Run 26

Calculated Film Boiling Data for n-Pentane on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
131.65	41017.8	311.56	4633.3	168.88	126.06	24.13	27.83
202.82	50672.7	249.84	5788.1	272.03	219.13	102.27	35.38
261.00	45019.9	172.21	7487.2	342.21	295.21	199.23	36.85
335.79	71882.6	214.07	9463.1	468.06	393.02	278.53	44.24
419.35	94485.2	225.31	11368.7	600.93	502.29	374.69	53.68

TABLE XXVII-A

Run 27

Calculated Film Boiling Data for Freon 113 on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

ΔT (°F)	Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Heat Transfer Coefficient $\frac{Btu}{Hr\ Ft^2\ °F}$	Thermal Heat Flux $\frac{Btu}{Hr\ Ft^2}$	Energy Input $\frac{Btu}{Hr}$	Energy Lost $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
136.44	(805.3)	(5.90)	8943.8	181.98	182.83	74.09	36.36
194.90	1710.8	8.78	7898.8	249.47	247.68	120.28	39.09
255.17	12680.6	49.69	7997.3	327.78	314.55	168.53	47.70
323.63	54209.9	167.51	9497.0	447.08	390.48	235.55	58.73
375.68	76567.6	203.81	10068.9	528.16	448.23	318.56	64.70

TABLE XXVIII-A

Run 28

Calculated Loss Data for n-Pentane on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

Core Temperature °F	Energy Input $\frac{Btu}{Hr}$	Auxillary Power Input $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
198.22	84.79	144.61	114.70	26.75
258.39	161.93	144.61	156.20	29.82
313.45	234.86	144.61	265.93	43.80
393.87	340.52	94.70	31.12	59.83
460.00	423.65	11.71	256.17	62.37
548.70	548.26	0.00	357.70	74.36

TABLE XXIX-A

Run 29

Calculated Loss Data for Benzene on a 0.4375 Inch
Diameter Surface at 14.7 psia Pressure

Core Temperature °F	Energy Input Btu/Hr	Auxillary Power Input Btu/Hr	Energy Gained by Cooling Water Btu/Hr	Energy Lost through Insulation Btu/Hr
300.22	198.05	192.78	73.64	35.01
349.09	253.09	76.08	115.09	82.15
407.91	318.05	45.49	131.27	88.84
463.63	379.12	7.89	142.27	94.99
506.43	423.65	0.00	205.99	94.6
553.35	477.57	0.00	261.45	99.33
577.17	518.88	0.00	300.06	101.28

TABLE XXX-A

Run 30

Calculated Loss Data for Freon 113 on a 0.4375 Inch Diameter Surface at 14.7 psia Pressure

Core Temperature °F	Energy Input $\frac{Btu}{Hr}$	Auxillary Power Input $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
197.30	122.13	84.22	73.34	37.08
230.87	154.57	39.59	62.07	41.12
287.55	218.60	2.32	59.50	49.01
338.82	276.85	0.00	104.24	56.11
403.55	344.79	0.00	176.25	64.43
458.22	408.30	0.00	227.48	71.85
514.39	466.18	0.00	310.11	79.43
553.74	510.39	0.00	352.94	82.73

TABLE XXXI-A

Run 31

Calculated Loss Data for Freon 11 on a 0.4375 Inch
Diameter Surface at 14.7 psia Pressure

Core Temperature °F	Energy Input Btu/Hr	Auxillary Power Input Btu/Hr	Energy Gained by Cooling Water Btu/Hr	Energy Lost through Insulation Btu/Hr
184.74	84.86	36.87	70.83	15.17
232.00	150.14	7.45	76.53	22.51
275.91	210.25	0.00	103.41	29.28
325.18	278.81	0.00	151.08	36.77
389.73	376.83	0.00	208.27	46.55
449.59	446.12	0.00	265.08	55.72
514.52	532.07	0.00	345.88	65.59
550.09	584.84	0.00	411.67	70.94

TABLE XXXII-A

Run 32

Calculated Loss Data for n-Pentane on a 3.630 Inch Diameter Surface at 14.7 psia Pressure

Core Temperature °F	Energy Input $\frac{Btu}{Hr}$	Auxillary Power Input $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
194.26	72.77	147.96	208.69	25.75
245.73	141.38	38.08	301.02	32.40
302.17	219.87	0.00	159.58	40.51
375.95	326.98	0.00	235.44	52.53
484.41	455.85	0.00	331.13	67.03

TABLE XXXIII-A

Run 33

Calculated Loss Data for Freon 113 on a 3.630 Inch Diameter Surface at 14.7 psia Pressure

Core Temperature °F	Energy Input $\frac{Btu}{Hr}$	Auxillary Power Input $\frac{Btu}{Hr}$	Energy Gained by Cooling Water $\frac{Btu}{Hr}$	Energy Lost through Insulation $\frac{Btu}{Hr}$
229.26	156.22	64.17	118.45	41.27
302.30	235.89	63.85	202.27	49.19
390.41	331.77	19.63	236.78	62.23
503.05	454.32	0.00	372.90	75.44